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PROCEEDINGS

"ENGINEERING PLANT LIFE ASSESSMENT"

M.A. IWANIW and S.D. BHOLE

EDITORS

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Dr. R. Seshadri
Dean, Faculty of Engineering
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PREFACE

The Eleventh Symposium on Engineering Applications of Mechanics was held in Regina in May, 1992. The general theme of the Symposium was "Engineering Plant Life Assessment", but technical papers addressing other areas of the application of mechanics were presented at the Symposium. Specifically, the role and application of mechanics pertaining to the areas of general design, materials and fabrication, mining, agriculture, energy and advanced technology were explored.

In accordance with the general aims of this Symposium series, the basic objective was the promotion of communication and transfer of technology in the field of mechanics between industry, governmental agencies and universities. In order to extend the dialogue and create further interest among participants, two keynote addresses, four professional development sessions and one panel session were included. All papers were reviewed within the University of Regina. The wide range of contributions of those participating in the Symposium was certainly encouraging.

The overall success of the Eleventh Symposium on Engineering Applications of Mechanics can be attributed to the effort of the Organizing Committee and the generous support of the sponsors - Atomic Energy of Canada Limited, IPSCO, Kalium Canada Limited, National Research Council, Saskatchewan Research Council, SaskEnergy, SaskPower and Syncrude Canada Limited.

R. Seshadri
W.B.H. Cooke
Co-chairs, 11th EAM Symposium
University of Regina

NOTE FROM THE EDITORS

The Proceedings of the 11th Symposium on Engineering Applications of Mechanics consists of the presentations at the two Plenary Sessions, the forty-seven presentations of the eight technical sessions, and the biographies of those involved with the Symposium. The majority of participants in the Symposium were from Canada, but there was strong representation from the United States, and some from as far away as Hong Kong and Rio de Janeiro. We were particularly pleased with the success of the Professional Development Sessions which allowed engineers in Canadian industry the opportunity to learn of state-of-art methods of risk and life assessment from acknowledged experts.

The Editors would like to acknowledge the assistance of the following people, who played a major role in the production of these Proceedings: our fellow faculty members at the University of Regina who critiqued the submitted papers; the staff of Duplicating Services at the University of Regina for their help in the design and layout of all published material pertaining to the Symposium; the graduate students of the Industrial Systems Engineering programme for their assistance in preparing mats; and Melody Murray, Nancy Bedard and Fran Mollard of the Engineering office for word-processing.

M.A. Iwaniw
S.D. Bhole

Editors

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PLENARY SESSIONS



"Life Extension and Component Condition Assessment"

Barry R. Dooley
Manager
Boiler and Cycle Chemistry Research
Electric Power Research Institute
Palo Alto, California

(text unavailable at time of printing)

CODE BOILER AND PRESSURE VESSEL CONDITION ASSESSMENT

James R. Farr, P.E.
Fellow-ASME

Babcock & Wilcox
a McDermott Company
Barberton, Ohio, U.S.A.

In this presentation, some of those considerations which are made in the United States of America and in Canada for deciding to do a condition assessment of boilers and pressure vessels designed and constructed to the ASME Boiler and Pressure Vessel Code are reviewed. Condition assessment or life assessment is essential in determining what is necessary for continued operation. With no ASME rules being adopted by laws or regulations, other than OSHA in the USA and similar environmental controls in Canada, to control condition or life assessment for continued operation, the equipment owner must decide if assessment is to be done and how much to do. Some of those considerations are reviewed along with methods and procedures to make an assessment along with a discussion of where the ASME B&PV Code currently stands regarding continued operation.

INTRODUCTION

In the United States of America and Canada, the design and construction of boilers and pressure vessels is according to the ASME Boiler and Pressure Vessel Code. The states and provinces adopt by law those sections of the ASME Code which are felt to be applicable to them. Not all Code sections are adopted by each "local jurisdiction."⁽¹⁾ The laws which have been adopted in each local jurisdiction determine which code sections are mandatory for design and construction of pressure components. Except for ASME Section III, Nuclear Components, the operation, inspection, evaluation, and repair or alteration of the boilers and pressure vessels is controlled by the local laws and requirements. For Section III pressure vessels, these items are controlled not only by the local jurisdiction but also by a federal agency (Nuclear Regulatory Commission, NRC, in USA and Atomic Energy Commission, AECL, in Canada). This control includes inspection and evaluation for life assessment, life extension, and continued operation using the requirements given in ASME Section XI, Nuclear Inservice

Inspection. On the other hand, there are no ASME Code rules for mandatory inspection and evaluation of boilers and pressure vessels built to any other sections of the ASME B&PV Code. Since the ASME Section III rules and concepts have been adopted not only in the USA and Canada, but also in many countries in the world, numerous papers and reports have been written and various interpretations have been established as to what is required for life assessment and continued operation of nuclear equipment to satisfy the laws and regulations. However, within the last few years, attention has turned, through discussions, meetings, and written documents, to the inspection and evaluation of boilers and pressure vessels for continued operation.

In general, the words: life assessment and life extension may not be as appropriate for boilers and pressure vessels as words such as: condition assessment, simply because many boilers and pressure vessels were not designed for a finite life of operation nor did the ASME Code require them to be designed for some finite life. For many older boilers and pressure vessels, condition assessment with the goal of continued operation is done by contractual agreement between companies and consultants who offered their experience and expertise to users and owners of the equipment.

In many instances, the timetable for inspection and evaluation is established by the outage

⁽¹⁾ Local jurisdiction is the municipal, state, provincial, or federal authorities enforcing boiler and pressure vessel laws or regulations issued by their legislature.

schedule of the equipment and by cost. In some instances, the inspection timetable is established by local laws which required a periodic issuance of a license or permit to operate the equipment. This inspection may be very superficial or it may be extensive. Since the inspection and evaluation of Section III nuclear pressure vessels is well documented, those requirements will not be discussed. This presentation will deal with the condition assessment for continued operation of boilers and pressure vessels built to the ASME Code, Sections I, IV, and VIII, and possibly API 620 and 650 vessels. It could also apply to other equipment.

WHY CONDITION OR LIFE ASSESSMENT?

Although the original designs of older boilers and pressure vessels usually did not contain specific lifetime of operation in their design specifications, designers made an attempt to provide a "long life" of operation by incorporating such features such as rounded corners, smooth ground welds, and other features to maintain low stress intensification factors; extra thickness was added to keep basic (primary) stresses low; and flexibility was incorporated into components to keep those stresses low. Ultimately, some of these features helped and some did not. Probably, the one single item which was most contributory to long life is the fact that operational temperature and/or pressure was well below design values.

Life assessment of boilers and pressure vessels which have been operating for some time are based on two criteria: safety and cost. Will a failure cause possible injury or death of plant personnel? Where plant safety is not in jeopardy, would failure cause an extended outage or result in damage to the equipment which would be too costly to repair? Would it be less costly to repair or replace the components on a scheduled basis, or is it less costly to wait until failure occurs in those components for which safety is not a concern?

As shown on Figure 1, taken from Steam 40 ⁽²⁾ and modified to consider both boilers and pressure vessels, the early life consists of many outages to get the equipment operating smoothly. After a short time, forced outages remain low; however, as life reaches 10 or 15 years for a pressure vessel, depending on the use, and 30 or 35 years for a utility boiler, forced outages begin to increase. In the past, that was the time that "end of life" of the equipment was considered. In today's atmosphere, it now has become a time for repair, replacement or scrap. One of the major causes of failures in boilers and pressure vessels is extensive cycling operation of the equipment which had little or no consideration of fatigue life in the original design of the component. This, along with erosion and corrosion, and operation above the specified temperature and pressure limits has led to component failure and expensive downtime. In order to help prevent failures of this type, condition assessments are made. As a result of

these assessments, repairs, alterations, and replacements may be made, or the equipment may be adjudged to be satisfactory for continued operation either at the original design conditions or at some lesser operating conditions. Condition assessment may also result in a change or restriction in operating procedures or in revised inspection schedule.

TYPES OF CONDITION ASSESSMENT

Basically, there are three types of condition assessment which are used to review and evaluate boilers and pressure vessels for continued operation. These three methods are visual examination, nondestructive and destructive examinations, and calculation and evaluation. Each of them may overlap the others in determining what is required for continued operation.

Visual Examination

The first method is visual examination. It is by far the simplest method and consists of the examiner looking at the equipment or component often aided by enhanced lighting, mirrors, borescopes, and magnification. Rough measurements of the thickness of components may be made in areas which are assessable. The examination is often supplemented by a review of the operating records, if available, and interviews of the plant operating personnel. For the experienced examiner, this often reveals areas of major concern which are visible (only on the outside surfaces of those components which can not be examined on the inside).

Nondestructive and Destructive Examinations

The second method of condition assessment is considerably more complex than a visual examination. It consists of nondestructive examination (NDE) where no damage of the equipment is desired nor made, and destructive examination of some parts and components where permitted or necessary. The two types of examinations are widely used in the boiler and pressure vessel industry.

(1) Nondestructive examinations. Most of the nondestructive examinations which are used to examine components during condition assessment are the same methods which are used to examine and inspect welds and other parts of components during fabrication as described in detail in ASME Code, Section V, Nondestructive Examination. Some of these methods of examination are: magnetic particle, liquid penetrant, ultrasonic, eddy current, radiography, and acoustic emission. There are a number of variations on some of these.

Magnetic particle testing (MT) detects surface and near surface indications in components which are made of ferromagnetic materials which can be magnetized. In MT examinations, iron particles are applied to the area being inspected and a magnetic field is set up in which the particles arrange in a pattern along the lines of magnetic force. An indication, crack, or flaw will interrupt these lines of force and cause a deviation in the pattern which can be easily observed.

⁽²⁾ From Steam 40, Courtesy of Babcock & Wilcox, see references

Liquid penetrant testing (PT) detects surface indications and cracking in components without regard to the magnetic properties of the material being tested. A liquid dye penetrant is applied to a clean surface being examined and allowed to dry. Some of the dye is drawn into the surface crack. A developer is then sprayed on to the dye and the crack becomes visible.

Ultrasonic testing (UT) is one of the quickest methods of NDE of a component. There are many variations for making an ultrasonic examination. UT can be used for detecting flaws, determining thickness of parts, of deposits, and of oxide layers. To conduct an ultrasonic test, a piezoelectric transducer is moved over the surface of the material being examined. This causes a disturbance in the atomic structure and permits sound waves to move through the material. The wave is reflected by the flaw and is picked up by a receiver where it can be viewed on an oscilloscope screen. The travel time for the wave varies with the distance; therefore, it is easy to examine the displaced signal to determine the flaw size and location, a metal thickness, or inner surface damage pattern. The PISC test program has shown that the personnel problem relative to interpretations of patterns is a far greater problem than the signals themselves.

Eddy current testing (ET) is accomplished by measuring the effects of induced eddy currents on the electromagnetic field using alternating current. The method can be used in any material which will conduct electricity. The patterns formed are compared against acceptance standards.

Radiographic testing (RT) is the major NDE method used during the construction of boilers and pressure vessels both in the shop and in the field. A radiograph is formed by the passing of x-rays and gamma rays through the material being examined. Some of the rays are absorbed depending upon the thickness and density. The rays which pass through are exposed to a film producing an image. Differences in density caused by a flaw will permit the rays to pass more quickly through that thickness and cause a darker image in that region on the film. Acceptance standards have been established to aid the "reader" to determine acceptability of the component. A special version of radiography is called Real-Time Radioscopic Examination which permits immediate viewing of the image on a screen instead of a film. This method has been known in the past as fluoroscopic examination. One problem is that there is no film to retain the image; and, consequently, the image has to be stored on tape if necessary.

Acoustic emission (AE) examination utilizes transducers to locate subsurface crack growth when stresses are generated from applied pressure or heating. Due to the stress concentration at the defect, localized yielding and plastic deformation take place which can be detected by the transducers. By taking the signals from several transducers, triangulation can locate the flaw.

Strain measurement is generally obtained by application of strain gauges to the surface of the metal. The gauges are either attached by adhesive or spot-welded to the component. The

gauge records the change in length by a change in resistance which is calibrated to a monitoring device. Special gauges are available for temperatures up to 1200F.

Temperature measurement is obtained using thermocouples applied at desired locations on the boiler and pressure vessel. Thermocouples can be attached to record temperatures on the surface or through the thickness.

(2) Destructive examinations. Destructive examinations are limited in use because they are more expensive to perform and require a repair or replacement of the component. This type of test is usually used when the physical properties (chemical and mechanical properties) of the material must be determined or when it is necessary to examine the inside of a component which is inaccessible from the outside. In a boiler, the most common component to be removed for destructive testing is a tube sample. However, in large components such as pressure vessels and steam drums, it may be necessary to remove a "boat" sample which is a wedge shaped piece removed from a larger component. The removed wedge then has to be repaired or replaced by welding.

Calculation and Evaluation

The third method of assessment involves calculations and evaluation of results. It is the most expensive and time-consuming and is generally used only when the visual examination and the nondestructive and destructive examinations are felt to be marginal in predicting continued operation needs. It is also used when equipment monitoring is needed to supply operating information for calculations and evaluation of results for continued operation and life extension without shut-down of the equipment.

Currently, design and construction of ASME boilers and pressure vessels follow the rules and requirements contained in ASME Code, Section I - Power Boilers,

Section IV - Heating Boilers, and Section VIII - Pressure Vessels. These sections of the Code follow the following format:

- (1) allowable design stresses for acceptable materials;
- (2) formulas, equations, charts, and tables along with illustrations of design details for safe construction;
- (3) acceptable forming, welding, and other fabrication methods;
- (4) inspection and testing methods;
- (5) safety valves and pressure relief equipment; and
- (6) nameplates and stampings with operational and design limits.

When all ASME Code requirements are met, the proper ASME stamp is applied. This information is basic to the further calculations and evaluations which are to be made for assessment of continuing operation.

Looking at the operational data records and physical information which has been obtained from the visual examination, NDE, and other testing, comparisons are made with the original thicknesses and sizes. In addition to the usual requirements of thickness, flaw detection, and erosion/corrosion, it may be necessary to determine methods for creep, fatigue, and creep-

fatigue interaction beyond those addressed in Code Case N-47 and for many additional materials. When that evaluation is completed, a decision is made to repair, replace, or scrap. Other evaluations are much more complex, especially when evaluations are made while the equipment is operating rather than during an outage. In these cases, data is obtained and fed into PCs and "black-box" monitors where computerized evaluations determine that a change in operating mode will enhance "life" with shut-down not required at that time.

WHERE DOES THE ASME BOILER & PRESSURE VESSEL CODE STAND?

Currently, except for ASME Section III and Section XI, there are no Code rules for condition or life assessment of operating equipment for continuing operation. The only sections of the Code which come close to addressing boilers during operation are Section VI - Recommended Rules for the Care and Operation of Heating Boilers and Section VIII - Recommended Guidelines for the Care of Power Boilers. Both sections are nonmandatory, but they do contain suggestions for operation and inspection which include checklists of those items in boilers which should be examined for a good start-up and continued operation. Briefly stated, the current ASME Code sections which have been adopted by the local jurisdictions cover the design and construction of new equipment. However, once that equipment is operated, there are no mandatory rules to cover it.

Section VIII, Division 2, Alternative Rules for Pressure Vessels, contains rules and requires the consideration of design life and fatigue cycles in the design of pressure vessels. However, ASME has no control of that vessel during its operational life. This is a real anomaly to include items in the design specifications and yet have all of the control of operating life rest with the owner. Currently, three ASME documents are being developed for issuance for the design and construction of high pressure vessels in the range of 10,000 psi to 100,000 psi. The three are high pressure systems, high pressure piping, and high pressure vessels. The high pressure vessel document is in its final stages of preparation for the first edition either as Section VIII, Division 3, or as some other ASME Code. The importance is that Section VIII-3 is a Section which permits only "design by analysis" but still avoids the issue of operation or life assessment for continued operation as an ASME controlled requirement.

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TECHNICAL SESSION 1
MATERIALS-DESIGN ANALYSIS I

Chair: M.C. Singh

FLANGED JOINT ANALYSIS: A SIMPLIFIED METHOD BASED ON ELASTIC INTERACTION

by

A. Bouzid*, A. Chaaban**

*Ph.D candidate, **Associate professor
Department of Mechanical Engineering
Ecole Polytechnique of Montreal
P.O. Box 6079, Station A
Montreal (Quebec)
H3C 3A7

ABSTRACT

Structurally sound bolted joints often fail due to loss of tightness. This is because the clamping load is affected by the application of the internal fluid pressure. A good design technique should therefore encompass most aspects of joint behaviour and produce efficient sealing performance within the clearly defined limits of the method used.

This paper presents a simple analytical model based on an extension of the Taylor Forge approach taking into account flange rotation, flexibility of both the gasket and the bolts and, when applicable, the stiffness of the end closure.

Examples will be discussed based on experimentally determined gasket properties.

NOMENCLATURE

a	- slope of tightness loading curve
A, in	- flange outside diameter
A _b , in ²	- bolt area
A _p , in ²	- pressure area
A _g , in ²	- full gasket contact area
b, in	- effective gasket seating width
B, in	- flange inside diameter
C, in	- bolt circle diameter
D, lb. in	- plate flexural rigidity
D ₀ , in	- diameter location of flange centroid
D _g , in	- gasket deflection
D _{max} , in	- maximum gasket deflection
E _b , psi	- bolt modulus of elasticity
E _c , psi	- cylinder modulus
E _f , psi	- flange modulus
E _g , psi	- gasket modulus
F _b , lb	- bolt force
F _g , lb	- gasket force
G, in	- effective gasket diameter
G _b , psi	- loading intercept on a tightness curve
G _g , psi	- unloading intercept on a tightness curve
K	- ratio outside to inside diameter
K _b , lb/in	- bolt uniaxial stiffness
K _g , lb/in	- gasket uniaxial stiffness
K _h , lb/in	- end closure uniaxial stiffness
l _b , in	- initial bolt length
m	- gasket factor
M ₀ , inlb/in	- discontinuity moment
M _f , inlb/in	- total equivalent flange moment

n	- number of bolts
N, in	- effective gasket width
N ₁ , lb/in	- longitudinal force in cylinder
P, psi	- internal pressure
P ₀ , lb/in	- discontinuity force
S _g , psi	- gasket stress
S _{o1} , psi	- gasket operating stress
S _{o2} , psi	- gasket seating stress
S _{ya} , psi	- theoretical gasket seating stress
t _c , in	- cylinder thickness
t _f , in	- flange thickness
t _g , in	- initial gasket thickness
T _p	- tightness parameter or factor
T _{min}	- minimum tightness
T _r	- tightness ratio
u _{1,2,g} , in	- axial displacement of gasket
u _b , in	- axial displacement of bolt
u _f , in	- axial displacement of flange
u _h , in	- axial displacement of end closure
V, lb	- frictional force
w _c , in	- radial displacement, cylinder
w _f , in	- radial displacement, flange
W _b , lb	- design bolt load
W _{o1} , lb	- operating design bolt load
W _{o2} , lb	- seating design bolt load
y, psi	- gasket design seating stress
Y	- factor involving K
ε _c , in ⁻¹	- shell constant
γ	- flange constant
θ _c , rad	- shell rotation
θ _f , rad	- flange rotation
ν _c	- cylinder Poisson's ratio
ν _f	- flange Poisson's ratio

INTRODUCTION

A number of questions have been raised for the past 10 years with regards to the precision of the methods used for the design of bolted flanged connections. Indeed, in the field, investigators showed that leakage in gasket joints is still a major problem yet to be overcome. With higher pressures and temperatures becoming evident due to the required increasing performance and efficiency in power and chemical plants, bolted joint connections require rigorous analysis for both structural integrity and leakage tightness. In most joint analysis, little attention is focused on leakage performance of the

joint while all emphasis is put towards structural integrity by keeping all joint component stresses below specified levels.

In the widely used procedure of the ASME code [1], an ambiguity exists as to the exact definition of what a "tight joint" is. Is it to be understood that to assure a tight joint means that one of the design condition is no leak?

Obviously, there will always exist leakage to some extent, especially in cases of gaseous joints; but the best design is the one that reduces leakage to a minimum acceptable level. With the new strict environmental regulations, manufacturers have to put more emphasis on the leakage performance of their gasketed joints. One of the first gasket manufacturers to introduce the concept of leakage in the design procedure of gasket joints is Klinger [2] who produced charts based on a leakage of 0.25 ml/min considered to be the limit of a technically tight seal. Recently, Bazergui et al [3] have successfully applied the concept of a tightness factor T_p that has been defined in Ref. [4,5] and used in their new proposed ASME like optimizing design procedure. The traditional m and y factors, introduced several decades ago [6], were substituted by new idealized constants G_b , a and G_g which have well characterized experimentally the performance of gaskets.

A first basic development is to find a relationship between the bolt initial seating and final operating loads taking into account flange rotation, flexibility of both the gasket and the bolts and, when applicable, the stiffness of the end closure. The present work is a preliminary study of this aspect of the problem. The philosophy adopted is similar to the one used in the early Taylor Forge method [7].

The proposed design equations will be used to analyze and discuss some practical cases.

ANALYSIS

Fig. 1 shows schematically two types of bolted flanged joint that have been considered in the present work. The following assumptions have been adopted throughout the analysis:

- The stress is uniform across the gasket width,
- The gasket compression is linearly related to its displacement and,
- The flange rotates rigidly with respect to a point of zero radial deflection located in the gasket-flange contact region.

For the purpose of the analysis a free body diagrams of the bolted joint is required. As an example a flange with an end closure joint (type 1) is shown in Fig. 2 for both the seating and operating conditions.

Within the framework of the previous considerations, a system of equations considering the force and moment equilibriums together with compatibility may be derived in the manner described in Ref. [8]. Based on Fig. 2, the axial equilibrium of external forces are obtained for the seating and operating conditions respectively:

$$F_b^i = F_g^i \quad (1)$$

$$F_b^f = F_g^f + P A_p \quad (2)$$

$$N_l = P \frac{B}{4} \quad (3)$$

The initial bolt load may be derived from the more modern method based on the design criterion of tightness. This method is the result of the extensive

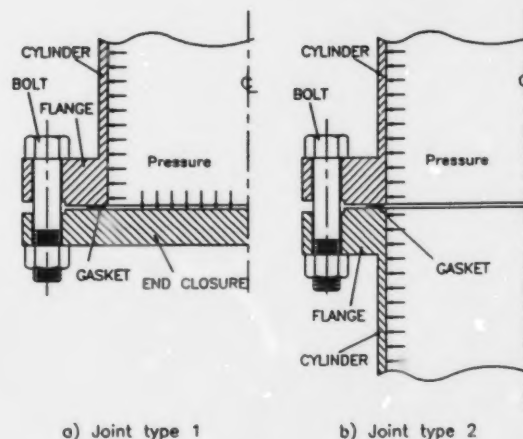


Figure 1 Examples of bolted flanged gasketed joint

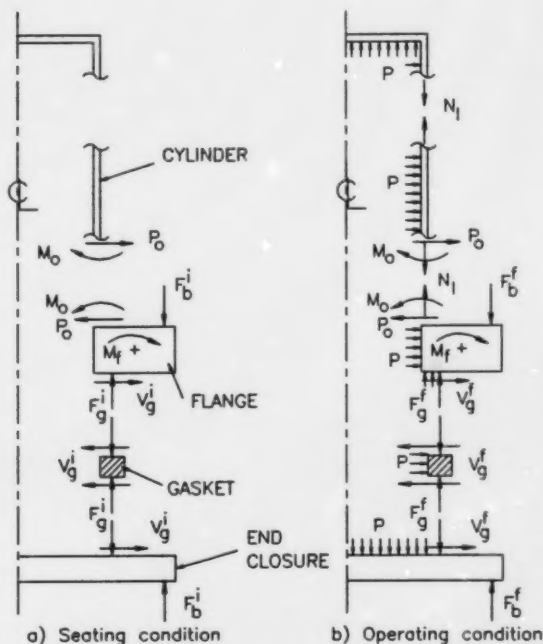


Figure 2 Free body diagrams of joint type 1

gasket leakage tests that have been carried out under the auspices of the Pressure Vessel Research Committee (PVRC) of the Welding Research Council (WRC) [9,10,11]. In order to optimize the bolt load, a similar but more balanced iterative procedure that uses a variable tightness criteria [12] was developed. The latter optimizes the values of the ASME Code-like seating and operating design loads (W_{a1} and W_{a2}) taking into consideration certain physical constraints on the joint. First, the minimum gasket stress should always be greater than $2P$ and secondly, the seating stress S_{ya} is reduced by 1.5 so as to avoid higher allowable stresses for the assembly condition.

Therefore, the minimum bolt load W_a is derived by satisfying both the seating and operating gasket stress requirements for the design value of minimum tightness T_{pmin} during operation, given a specified design

pressure, gasket geometry and constants G_b , a and G_s , such that;

The required theoretical seating stress, S_{ys} is:

$$S_{ys} = \frac{G_b}{0.75} [1.5 T_{min}]^2 \quad (4)$$

The design seating stress component, S_{s2} is:

$$S_{s2} = \frac{S_{ys}}{1.5} - P \frac{A_p}{A_s} \quad (5)$$

The design operating stress component, S_{s1} is:

$$S_{s1} = G_s \left[0.75 \frac{S_{ys}}{G_s} \right]^{1/2} \quad (6)$$

where $T_r = \log(1.5 T_{min}) / \log(T_{min})$

The minimum bolt load required, W_m , is:

$$W_m = P A_p + S_m A_s \quad (7)$$

S_m is the greater of S_{s1} or S_{s2} or $2P$.

But first let us consider the cylindrical part of the assembly. The cylinder end is subjected to the discontinuity edge moment M_0 and shear force P_0 per unit circumference. The radial displacement w_c and slope θ_c at the junction are given by the theory of beam on elastic foundation [13]:

$$w_c = \frac{6(1-\nu_c^2)P_0}{E_c t_c^3 \beta_c^3} - \frac{6(1-\nu_c^2)M_0}{E_c t_c^3 \beta_c^2} + \frac{(2-\nu_c)B^2 P}{8E_c t_c} \quad (8)$$

$$\text{where } \beta_c = \left[\frac{12(1-\nu_c^2)}{B^2 t_c^3} \right]^{1/4}$$

$$\text{and } \theta_c = \frac{12(1-\nu_c^2)M_0}{E_c t_c^3 \beta_c} - \frac{6(1-\nu_c^2)P_0}{E_c t_c^3 \beta_c^2} \quad (9)$$

The radial displacement of the flange w_r at the junction with the cylinder may be calculated by considering the thick-walled cylinder theory [14]. The total displacement is given by:

$$w_r = \frac{B\gamma P}{2E_r} - \frac{B\gamma P_0}{2E_r t_r} + \frac{\gamma V_s}{2\pi t_r E_r} + \frac{\theta_r t_r}{2} \quad (10)$$

$$\text{where } \gamma = \frac{A^2 + B^2}{A^2 - B^2} + \nu_r$$

There are four terms in Equation (10), the first one is due to the internal pressure P , the second and third are due to the discontinuity edge force P_0 and the friction force V_s which is considered uniformly distributed over the thickness of the flange, and the last term is due to the flange rotation about its centre of gravity. The flange rotation θ_r may be obtained from the theory of a circular plate with a central hole subjected to an equivalent circumferential twisting couple M_r located at the inside and outside diameters of the flange as detailed in ref. [15], [16]:

$$\theta_r = \frac{B\pi Y M_r}{E_r t_r^3} \quad (11)$$

$$\text{where } Y = \frac{1}{K-1} \left[\frac{3}{\pi}(1-\nu_r) + \frac{6}{\pi}(1+\nu_r) \frac{K^2 \ln K}{(K^2-1)} \right]$$

with $K = A/B$

Referring to Fig. 2, the twisting couple M_r can be obtained from considering the equilibrium about the flange centroid as;

$$M_r = -\frac{B}{D_0} M_0 - \frac{B}{D_0} \frac{t_r}{2} P_0 + \frac{C}{D_0} \frac{(D_0-G)}{2} \frac{F_s}{\pi C} + \frac{C}{D_0} \frac{(C-D_0)}{2} \frac{F_b}{\pi C} - \frac{G}{D_0} \frac{t_r}{2} \frac{V_s}{\pi C} + \frac{(G-B)}{4} P + \frac{\pi}{4} \frac{(G^2-B^2)}{\pi D_0} \frac{(D_0-G)}{2} + \frac{B}{D_0} \frac{(D_0-B)}{2} N_1 \quad (12)$$

Substituting for F_s and N_1 from Equations (2) and (3), the expression for M_r from Equation (12) will reduce to:

$$M_r = -\frac{B}{D_0} M_0 - \frac{B t_r}{2 D_0} P_0 + \frac{(C-G)}{2 \pi D_0} F_b - \frac{t_r}{2 \pi D_0} V_s + \frac{(G-B)}{16 D_0} (G^2 + B^2) P \quad (13)$$

The equations for compatibility requiring continuity of both, rotation and displacement, Equations (8) and (10), also Equations (6) and (11) can be assembled together with Equation (13) to give a system of three equations with five unknowns; M_0 , P_0 , V_s , θ_r and F_b . Depending on the seating or operating condition, one or two additional equations are needed to solve for the unknowns. In fact, before application of pressure, the bolt initial force F_b^i , which is the required pretightening sufficient to satisfy both the requirements of seating and operating conditions (W_m and W_m^o), should be previously defined and consequently our system will be reduced to four unknowns. However, on application of pressure, the force in the bolt will obviously change to an unknown final value F_b^f requiring an additional fifth equation to solve the system.

The fourth equation is derived from considering the radial displacement of the pivoted point located in the flange and gasket contact surfaces. In the Taylor Forge method it is assumed that the radial displacement is zero at the thick end of the hub to reduce the number of unknowns and shorten the arithmetics. Clearly the resultant shear force P_0 and the frictional force V_s ; the latter not being introduced in the calculations; are the most important loads in the evaluation of the radial and hoop stresses in the flange.

In our model, a condition which will enable us to put into relation the quantities P_0 , V_s and θ_r comes from the pivoted point consideration referred to as point A in Fig. 3. The radial displacement of the flange w_r due to the edge loads P_0 and V_s must equal the radial displacement w_r of the flange produced by the flange rotation due to M_r if the static coefficient of friction is important enough to prevent radial displacement or slip at the gasket contact area.

$$w_r = w_r \quad \text{or} \quad \frac{\gamma}{2E_r} \left(\frac{V_s}{\pi t_r} - \frac{B P_0}{t_r} + B P \right) = \theta_r \frac{t_r}{2} \quad (14)$$

However, once some movement occurs, the static coefficient of friction is replaced by its dynamic value and the frictional force V_s would be simply; $V_s = \mu_{dyn} F_b$. The static and dynamic coefficients of friction may be obtained experimentally.

The still unknown final bolt load F_b^f for the operating condition may be obtained from the geometric compatibility consideration. The displacement of the nut Δ_n must be the same in the initial tightening state (1) and after the final pressurisation state (f); Fig. 3 shows schematically all displacement components taken into account for the geometric compatibility equation which is given by;

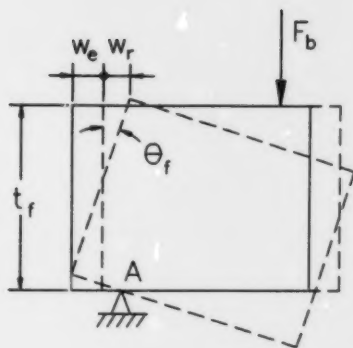


Figure 3 Flange pivoting around the contact zone

$$\Delta_o = u_b^i + u_b^f + u_b^i + u_b^f + u_b^i + u_b^f + u_b^i + u_b^f \quad (15)$$

The axial displacement u of the different joint members are obtained as follow:

a) The force-displacement relationships for the assumed linear behaviour of the gasket and bolt can be written as follows;

$$\text{for the gasket} \quad u_b = \frac{F_b}{K_g} \quad (16)$$

$$\text{with } K_g = \frac{A_g E_g}{L_g} \quad \text{and } A_g = \pi G N$$

In reality most gaskets show a very high non-linear behaviour and there is no distinct elastic range in their stress-strain curves as may be seen from part A of the graph in Fig 4, which is referred to the seating of the gasket. However, their loading and reloading portion of the graph, referred to as part B, show an almost linear variation typically present in most gaskets. From the slope of this latter part of the curve, a modulus of decompression, E_d , may be determined as in ref. [17];

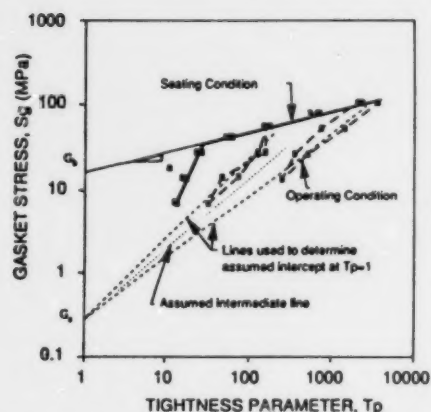
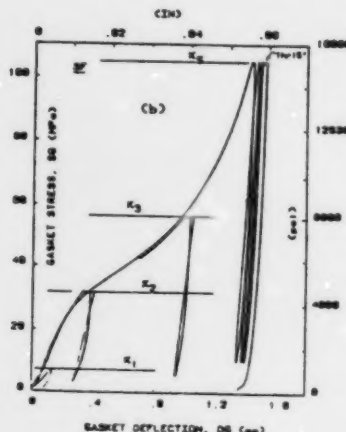
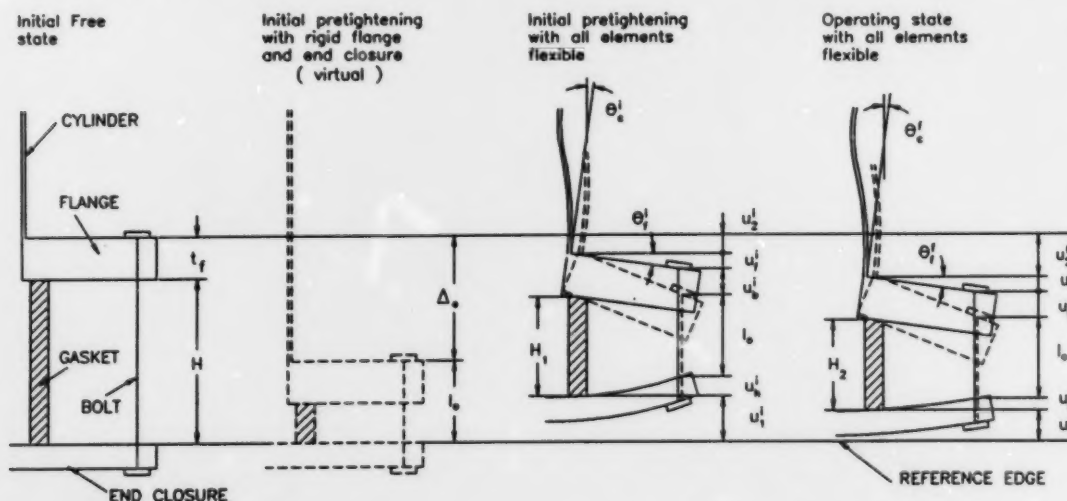


Figure 5 Typical gasket stress-deflection and tightness graphs, Ref. [4,12]



$$l_0 + \Delta_o = l_0 + u_b^i + u_b^f + u_b^i + u_b^f + u_b^i + u_b^f + u_b^i + u_b^f$$

$$\text{or } u_b^i = u_b^f + u_b^i + u_b^f \quad \text{and} \quad u_b^f = u_b^i + u_b^f + u_b^i + u_b^f$$

$$\text{then } \Delta_o = u_b^i + u_b^f + u_b^i + u_b^f = u_b^f + u_b^i + u_b^f + u_b^i$$

Figure 4 Different states including joint element displacements

$$E_s = \frac{dS_s}{dD_s} \frac{1}{(C_s - D_{smax})}$$

and for the bolt

$$u_b = F_b / K_b \quad (17)$$

$$\text{with } K_b = \frac{nA_b E_b}{l_b}$$

b) The Flange axial displacement u_f , as can be seen from Fig. 5, may be obtained from the rotation of the flange around the gasket pivot point such that:

$$u_f = \left(\frac{C-G}{2} \right) \theta_f \quad (18)$$

c) An equivalent system to the end closure shown in Fig. 6 is considered for a more precise evaluation of its axial displacement. This is achieved by the superposition of a flexible circular plate simply supported at the flange outside diameter and subjected to a ring load F_g distributed along the effective gasket diameter G , and an operating pressure P distributed over the inner part of the plate bounded by the same circle of diameter G and a similar plate subjected to a ring load F_b distributed along the bolt circle diameter C . Then, the components of displacement due to F_g , P and F_b given in [18] are:

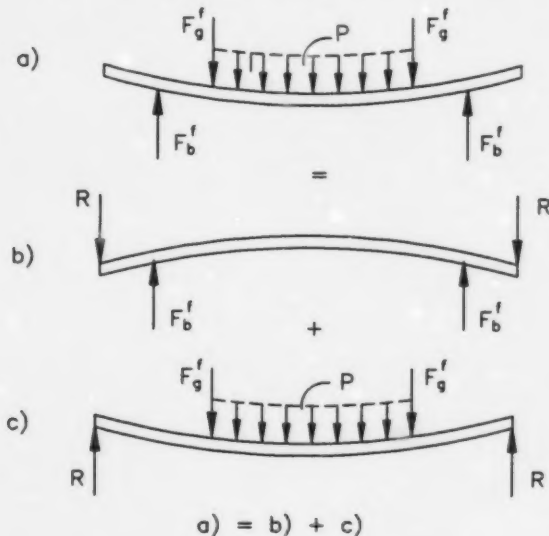


Figure 6 Equivalent end closure system

$$u_h = u_h(F_g) + u_h(P) - u_h(F_b) \quad (19)$$

$$\text{where } u_h(P) = a_0 P \quad (20)$$

$$a_0 = \frac{A^4}{1024D} [1 + 4 - 5\alpha^2 + 4(2 + \alpha^2) \ln \alpha] \alpha^2 + 2 \left(\frac{k_3}{1 + \nu} \right) C_1 - C_0$$

$$\text{and } u_h(F_g) = F_g / K'_h \quad (21)$$

$$\text{with } K'_h = \frac{64\pi D (1 + \nu_p)}{[k_1 - k_2 + k_2 C_1]}$$

where $\alpha = G/A$; $C_0 = 1 - \alpha^2$; $C_1 = 1 - \alpha^4$
The quantity D is the plate flexural rigidity;

$$D = \frac{Et_p^3}{12(1 - \nu_p^2)} \quad (22)$$

The constants k_i are defined as:

$$\begin{aligned} K_1 &= (3 + \nu)(1 - \alpha^2) + 2(1 + \nu)\alpha^2 \ln \alpha \\ K_2 &= (1 - \nu)(1 - \alpha^2) - 2(1 + \nu) \ln \alpha \\ K_3 &= [4 - (1 - \nu)\alpha^2 + 4(1 + \nu) \ln \alpha] \alpha^2 \end{aligned} \quad (23)$$

Similarly, the expression for $u_h(F_b)$ may be obtained by replacing F_g by F_b and K_h by K'_h , which is obtained by replacing $\alpha = C/A$ instead of G/A in all previous involved expressions

After assembling the above equations, it can be shown that the final bolt load may be given by the following equation;

$$\begin{aligned} (F_b)_f &= (F_b)_i + \frac{\left(\frac{1}{K_g} + \frac{1}{K'_g} - \frac{a_0}{A_p} \right)}{\left(\frac{1}{K_g} + \frac{1}{K'_g} + \frac{1}{K'_b} - \frac{1}{K'_b} \right)} A_p P \\ &+ \frac{(C-G)}{2 \left(\frac{1}{K_g} + \frac{1}{K'_g} + \frac{1}{K'_b} - \frac{1}{K'_b} \right)} (\theta_f^i - \theta_f^f) \end{aligned} \quad (24)$$

In the case of a symmetrical bolted joint (type 2) of Fig. 1, only half of the joint system including half of the gasket as well as half of the bolt may be considered. The approach is similar to the previous case but with the end closer considered as rigid. In general, the gasket being less rigid than the flanges; the whole gasket-flange system is equally shifted radially by the application of the pressure and the pressure term in equation (10) vanishes reducing to;

$$\frac{\gamma}{2E_f} \left(\frac{V_g}{\pi C_f} - \frac{BP_0}{C_f} \right) = \theta_f \frac{C_f}{2} \quad (25)$$

The same approach is used for the type 2 bolted joint to find the expression for the final bolt load and is given by;

$$(F_b)_f = (F_b)_i + \frac{\left(\frac{1}{K_g} \right)}{\left(\frac{1}{K_g} + \frac{1}{K'_b} \right)} A_p P + \frac{(C-G)}{2 \left(\frac{1}{K_g} + \frac{1}{K'_b} \right)} (\theta_f^i - \theta_f^f) \quad (26)$$

By neglecting the flexibility of the flange and end closure, the discontinuity force and moment at the cylinder to flange junction and the gasket to flange frictional resistance, all terms including K_h as well as θ are to be dropped out from the previous expressions of the final bolt load, leading to the same well known rigid flange expression;

$$(F_b)_f = (F_b)_i + \frac{1}{1 + K_g/K_b} A_p P \quad (27)$$

DISCUSSION AND CONCLUSION

A computer program, based on the elastic interaction model described in this paper and consistent with the ASME code approach, has been developed and implemented to give a better and more accurate estimation of the tightness and final joint loads. Based on an initial bolt loading obtained from the tightness or leakage concept using the new gasket factors G_b , a and G_s , the final bolt and gasket loads and stresses are obtained for the two types of flanges considered.

We consider three cases wherein two standard intermediate and large ANSI B16.6 class 1500 flanges [12] as well as a relatively large diameter flange, are

used either in pairs or with end closures in conjunction with three pressurized pipes NPS 12, 24 and 42 inches. The flange thicknesses are 4.88, 8 and 6.5 inches respectively and the flange material is carbon steel SA 105 of Young's modulus equal to 29.9×10^6 psi throughout. The design pressures are 1500 psi for the two ANSI B16.6 flanges cases and 925 psi for the large diameter flange case and the design temperature is 70 °F in all cases. Appropriate size double jacketed mica-filled gaskets with $C_g = 2900$ psi, $\alpha = 0.23$ and $G_s = 14.69$ psi with a compression modulus $E_g = 234$ ksi were used in all cases.

Table 1 summarizes the results obtained for the bolt and gasket loads based on elastic interaction of all joint elements. As shown in the table, contrary to the traditional ASME method, this approach gives a better evaluation of the final bolt and gasket loads as a function of the initial bolt loads, taking into account the flexibility of the gasket, the bolts and the flanges. With this method, the bolt and gasket stresses are also evaluated as shown in Table 2.

The final forces in the bolts and gasket obtained with the proposed equations are found to be higher than those predicted by the rigid flange method in all cases. An estimation of the frictional force existing between the flange and the gasket contact area is also given. In one hand, this force, being of the same order of magnitude as the gasket load, will clearly superimpose the effect of internal pressure increasing the flange stresses. From the other hand, it offers a resistance to the flange rotation and therefore decreases the flange stresses. This effect is more important in case of metallic gaskets having higher resistance.

Another point worth mentioning is that the flange rotations do not exceed 0.2° in all cases. In the case of the ANSI B16.6 class 1500 flanges, the rotation are rather low, less than 0.08° , owing to the fact that these flanges are rather stiff.

Table 1 Gasket and bolt loads.

Flange size NPS, in.		NPS 12	NPS 24	NPS 42
Design pressure, psi		1 500	1 500	925
Maximum bolt load, lb		472 232	1 396 700	1 900 000
Design bolt load W_g , lb		309 686	970 824	1 898 456
Final gasket load, (rigid flange), lb		90 598	234 513	492 245
Final bolt load, lb (new method)	Type 1	420 000*	1 278 000*	2 300 090*
	Type 2	426 600	1 314 000	2 354 000
Final gasket load, lb (new method)	Type 1	371 800*	1 082 000*	2 052 000*
	Type 2	396 200	1 166 000	2 310 000
Friction force operation, lb	Type 1	200 900*	541 600*	886 200*
	Type 2	207 500	578 200	940 400
Maximum flange rotation, deg	Type 1	152 700*	345 200*	638 300*
	Type 2	177 100	429 700	895 700
	Type 1	367 100	1 203 000	1 052 000
	Type 2	409 800	1 237 000	1 349 000
	Type 1	0.054*	0.074*	0.192*
	Type 2	0.031	0.041	0.100
	Type 1	0.049*	0.064*	0.178*
	Type 2	0.025	0.032	0.071

* with radial slip

Table 2 Gasket and bolt stresses.

Flange size NPS, in		NPS 12	NPS 24	NPS 42
Design pressure, psi		1 500	1 500	925
Initial gasket seating stress, psi		9 638	12 360	20 880
Final gasket stress, psi	Type 1	6 251*	6 896*	9 747*
	Type 2	6 459	7 361	10 340
	Type 1	6 251*	4 396*	7 021*
	Type 2	6 459	5 471	9 851
Initial bolt stress, psi		6 161	6 307	24 900
Final bolt stress, psi	Type 1	8 355*	8 302*	30 510*
	Type 2	8 487	8 539	31 220
	Type 1	7 396*	7 026*	27 220*
	Type 2	7 882	7 574	30 630

* with radial slip

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MODELLING OF MOLECULAR DIFFUSION COEFFICIENT IN DENSE BENTONITE

S.C.H. Cheung* and A. Mouradian**

*Associate Professor and ** Research Assistant
Civil Engineering Department, Concorida University
Montréal, Québec, Canada H3G 1M8

and

C.F. Lee

Manager

Civil Engineering & Architecture Department
Ontario Hydro
Toronto, Ontario
Canada M5G 1X6

ABSTRACT

To assess the safety of the disposal site, molecular diffusion coefficient through the bentonite-clay barrier is required. The methods commonly used to determine molecular diffusion coefficient in clay are very time consuming. Because of the large number of species involved in the radioactive waste disposal site, a model that allows diffusion coefficient to be predicted for use is desirable. Models based on free water have been proposed but are found to be inadequate for compacted bentonitic clays. A model that incorporates clay-species-water interaction is presented for compacted bentonite. The modelling results show that the diffusion coefficient depends on the charge nature and size of the diffusing species, soil density, water chemistry, temperature and soil structure. The predicted diffusion coefficients for some species are shown to be in excellent agreement with those measured in compacted bentonite.

INTRODUCTION

A major environmental problem is the intrusion of toxic contaminant from a waste disposal site into the groundwater supply. Bentonitic clays are often used as barriers to limit the movement of contaminants. An example of this is the compacted bentonitic clay known as buffer proposed for the Canadian Nuclear fuel waste disposal vault to surround the waste container in the borehole (Hancox, 1986) as shown in Figure 1. The permeability of this material is very low and the principal transport mechanism will be by molecular diffusion (Cheung and Gray, 1990). To assess the safety of the disposal site, molecular diffusion coefficient through the clay barrier is one of the critical parameters required. Methods have been developed to determine diffusion coefficient, but these methods are very time consuming. The measured diffusion coefficient is found to depend on density, soil structure and environmental factors such as porewater chemistry and temperature (Cheung 1989, Cheung 1990, Eriksen et al., 1986, Robin et al. 1987, Oscarson et al. 1992). Because of the large number of species involved (Hancox, 1986), the possible change of the barrier density due to its potential expansion into the tunnel backfill and rock fissures and the changing environmental conditions in the radioactive disposal vault, it is not feasible to cover the whole range of diffusion coefficients required. A model that allows the diffusion coefficient to be predicted based on readily obtainable parameters would be desirable. Models based on free water have been developed to predict diffusion coefficient in soil (Robin et al., 1987 and Manheim, 1970) but are found to be inadequate particularly for bentonitic clay (Cheung 1990). In this paper, we will discuss first the methods generally used to determine diffusion coefficient, the model based on free water for soil and then develop a model to incorporate clay-species-water interaction for bentonitic materials.

MEASURED DIFFUSION COEFFICIENT

In soil-water system, diffusion depends not only on molecular diffusion in the aqueous phase and physical chemical process such as sorption, ion-exchange and precipitation but also on pore size and tortuosity. Therefore, the measured diffusion coefficient is not a pure molecular diffusion in solution, and is generally referred to as the apparent diffusion coefficient.

Two methods based on Fick's law diffusion namely, in-diffusion and through-diffusion generally used to determine diffusion coefficient in soil are described below.

In-diffusion

In this method, the diffusion coefficient is determined from the concentration profile of soil. One common type of in-diffusion is shown in Figure 2 where a predetermined amount of soil is mixed with an appropriate amount of solution, with and without tracer. The two soils are then separately compacted to the same density. The soil water is allowed to equilibrate before the two soils are brought in contact. The tracer concentration profile is analyzed after a period of time necessary for diffusion.

For the initial and boundary conditions, the concentration profile is given by Eq. (1), provided that the tracer does not penetrate completely through the soil specimen (Crank 1975):

$$\frac{C_i}{\bar{C}} = \operatorname{erfc} \frac{x}{2(D_A t)^{1/2}} \quad (1)$$

where C_i is the tracer concentration at a distance, x and time t , $\bar{C}$ is the initial tracer concentration and D_A is the apparent transient-state diffusion coefficient.

Through-diffusion

In this method, one end of the water-saturated soil specimen is in contact with a constant tracer concentration solution, C_0 , while at the other end, the solution is tracer-free. The change in tracer concentration as a function of time in the initially tracer-free solution is measured. This provides an estimate of the cumulative amount of tracer diffused through the soil with thickness, d . The constant diffusive flux, F_{SS} with time is then used to calculate the apparent steady-state diffusion coefficient D'_A as

$$D'_A = F_{SS} d / C_0 \quad (2)$$

The apparent diffusion coefficients determined from the above methods will be used to compare with those predicted by the models described below.

FREE-WATER MODEL

The most fundamental diffusion coefficient is the free-water coefficient, D_0 , which links the diffusive flux, $\bar{F}$, to the concentration gradient in free water, ∇C_f , by Fick's first law:

$$\bar{F} = -D_0 \nabla C_f \quad (3)$$

For this model, Eq. (3) is retained in superficially the same form. However, the time-independent flux in the pore water per unit surface area of the soil, F_{SS} rather than the flux per unit surface area of bulk water is used to link the free water ∇C_f by the following coefficients:

$$F_{SS} = -D_p \theta \nabla C_f \quad (4)$$

where:

$$D_p = D_0 \tau \quad (5)$$

and D_p is defined as the overall pore diffusion coefficient, θ the water-saturated porosity and τ the tortuosity factor to account for the effect of tortuous diffusion path.

The conservation equation can be written as:

$$\theta \partial C_f / \partial t + \gamma_d \partial S / \partial t = \nabla \cdot (D_p \theta \nabla C_f) \quad (6)$$

In Eq. (6), γ_d is the dry density of soil and S denotes the amount of species sorbed per unit mass of dry soil which is related to:

$$S = K_d C_f \quad (7)$$

where K_d is the batch distribution coefficient, defined as the ratio of the amount of species sorbed per unit mass of dry soil to the concentration of that species remaining in solution, C_f , at equilibrium. Since K_d is determined in soil dispersed in solution, it implies that the soil particle surface areas are totally available to interact with the species. Assuming that the sorbed species are immobile and substitutes Eq. (7) for S into Eq. (6), the following Eq. is obtained

$$\partial C_f / \partial t = \frac{D_p}{1 + K_d \gamma_d / \theta} \nabla^2 C_f \quad (8)$$

In Eq. (8), D_p is retarded by a factor of $K_d \gamma_d / \theta$ because of sorption. A diagram illustrating this model is shown in Fig. 3.

In this model, sorption is assumed to be instantaneous and can be expressed in terms of a constant K_d . These requirements are likely to be met in the radioactive waste disposal site (Oscarson et al. 1989). This model has been previously used to predict and interpret diffusion coefficient in soil (Robin et al. 1987), and is attractive for use as the parameter, K_d is readily obtained from batch studies and τ (in Eq. (5)) is shown to be equal to θ for some soils (Manheim, 1970). For this model to be valid, the following experimental results would be expected.

- (1) D_A as described by Eq. (8) should not change with time,
- (2) D_A should be smaller for species with greater K_d , and
- (3) D'_A described by Eq. (4) for both anions and cations should not be much different since D_0 for these species are approximately the same and $\tau = \theta$ for both species.

This model has been found to be valid for less active soils (Robin et al. 1987 and Manheim 1970) but is not valid for compacted bentonite as shown by the experimental results in Fig. (4) and Table 1. An estimate of τ can be made from D_A/D_0 for anions from table 1 using $D_0 = 10^{-9}$ m/s. This value is found to be about 0.01 which is much smaller than θ (approximately 0.4). The lack of agreement is due to the effect of clay-species-water interaction which has not been taken into account in the model.

Clay-Species-Water Interaction Model

In this model, diffusion in the clay water is considered to be heterogeneous. Heterogeneous diffusion arises from the electrochemical forces between clay particles, clay water and dissolved species, and results in non-uniform distributions in ionic concentration and water viscosity (Cheung, 1989). The clay particles are generally negatively charged, and they attract cations and repel anions. As a result, the ionic concentration gradient along the clay-particle surface compared with that of free solution is higher for cations and smaller for anions. If the sorbed or attracted species are mobile, diffusive flux increases directly with higher concentration gradient. Moreover, the viscosity of water in the first two layers adjacent to the clay particle surface is higher than the rest of the soil water, which is not significantly different from that of pure solution. Diffusivity is inversely proportional to viscosity.

In developing the model for compacted bentonite systems (density > 1.2 Mg/m³), two main types of water, namely, surface and interstitial water, will be considered. Surface water with an average thickness up to 1 nm is assumed to form between parallel clay plates. The small surface-water thickness, when compared to the size of diffusing species, can restrict diffusion. Moreover, the viscosity and ionic concentration in this water influenced by the clay particles are significantly different from free water, as discussed earlier. Interstitial water is assumed to have the same properties as free water, and it forms predominantly in the larger pores of the compacted clay. A diagram illustrating this model is shown in Fig. 5.

For this model, F_{SS} for the two types of soil water is:

$$F_{SS} = -D_f \theta_f \nabla C_f - D_s \theta_s \delta_s \nabla C_s \quad (9)$$

where D , θ , δ , and C is the respective pore diffusion coefficient, the fractional volume of water, the restrictivity factor

for diffusion and the concentration for each type of water with subscripts f and s denoting free (interstitial) and surface water, respectively.

In these systems, surface water predominates, θ_s varies from 0.1 to 0.4 and θ_f is about orders of magnitude smaller than θ_s . The predominant diffusion pathway depends on θ_s , θ_f , δ_s , D_s , D_f , ∇C_f and ∇C_s .

Considering the dominant diffusion path in interstitial water, Eq. (9) can be written as.

$$F_{SS} = -D_f \theta_f \nabla C_f \quad (10)$$

where

$$D_f = D_0 \tau_f \quad (11)$$

This is likely to be the case for anions as they will be totally extruded from surface water resulting in $\nabla C_s = 0$ and also for the case where $\delta_s = 0$.

In Eq. (10), δ_f is taken to be 1 for small species as the pores are relatively large. For large species, some of the pores may not be large enough to allow the species to penetrate and δ_f needs to be considered.

Considering the dominant diffusion path in surface water, this is likely to be the case for cations and neutral species where $D_s \delta_s \nabla C_s > D_f \nabla C_f$. ∇C_s depends on the mobility of the adsorbed species.

For mobile cations, assuming linear ∇C_s , $\nabla C_s = K_d \gamma_d \nabla C_f$ and Eq. (9) becomes:

$$F_{SS} = -D_s \theta_s \delta_s m_i \nabla C_f \quad (12)$$

$$D_s = D_0 \tau_s \mu_s \quad (13)$$

and

$$m_i = K_d \gamma_d$$

where m_i is a factor to account for the increased concentration gradient due to cation exchange of mobile species and μ_s is the viscosity factor for surface water.

For immobile cations or neutral species, m_i is equal to 1 and Eq. (12) becomes:

$$F_{SS} = -D_s \theta_s \delta_s \nabla C_f \quad (14)$$

The conservation equation for the dominant diffusion pathway in surface water can be derived as (Cheung 1990):

For mobile cations from Eq. (12),

$$\frac{\partial C_f}{\partial t} = D_s \nabla^2 C_f \quad (15)$$

For immobile cations and neutral sorbed species from Eq. (14),

$$\frac{\partial C_f}{\partial t} = \frac{D_s}{1 + K_d \gamma_d / \theta} \nabla^2 C_f \quad (16)$$

The conservation equation for the dominant diffusion pathway in interstitial water can be derived as (Cheung 1990):

For non-sorbed species ($K_d = 0$),

$$\frac{\partial C_f}{\partial t} = D_f \nabla^2 C_f \quad (17)$$

For immobile sorbed species ($K_d > 0$),

$$\frac{\partial C_f}{\partial t} = \frac{D_f}{1 + K_d \gamma_d / \theta} \nabla^2 C_f \quad (18)$$

The above equations from (9) to (18) show that diffusion coefficient depends on the diffusion pathway, the charge nature of species, the relative sizes of soil pore and diffusing species, the dry density, the pore water chemistry and temperature (the type of species formed, K_d , θ_f and μ_s are affected by water chemistry and temperature, (Cheung 1990, Oscarson et al. 1989)).

For this model to be valid, the following experimental results would be expected.

- (1) D_A may vary with time - this is reflected by the initial decrease of D_A for the bentonite clays in Figure 4 which represents the averaging effect of diffusion in interstitial and surface water. The constant value of D_A with time indicates the dominant effect of diffusion in surface water. This effect is not shown in kaolinites as these systems are dominated by free water.
- (2) D_A can be smaller for cations than anions due to higher viscosity ($\mu_s < \mu_f = 1$) if $\tau_s = \tau_f$
- (3) D_A for mobile cations should not be significantly affected by K_d .
- (4) D_A for mobile cations can be greater than anions.

The experimental results in table 1 are shown to agree qualitatively with items (2) to (4) predicted by the model. The results also suggest that the exchangeable cations such as CS^+ weakly held by the clay particles, are mobile under the induced gradient. The restrictivity factor, δ_s is equal to $(K_d)_m / K_d$ where $(K_d)_m$ is the distribution coefficient in compacted soil which can be determined from the concentration profile of soil using the flow-through methods. Experiments have been conducted in compacted bentonites of 1.2 Mg/m³ for this purpose (Cheung and Gray 1989). The results show that the concentration profiles are linear and δ_s varies from 0.2 to 1 due to non-uniform distribution of pores between particles which exists between similar specimens. Moreover,

$F_{ss}/\theta_s \delta_s K_d \gamma_d \nabla C_f = D_s$ is found to be constant. This gives further support of the mobile nature of CS^+ . An estimate of θ_f has also been investigated from D_A in these systems using anions (Cheung and Gray 1989) and D_A from in-diffusion method employing Eq. (10). θ_f is shown to vary from 10^{-2} to 10^{-4} between similar specimens which is attributed to soil fabric effect.

For complexing species, such as $Pu(OH)_4$, they are strongly bonded to the clay particle and are not expected to be sheared under the induced concentration gradient. Using Eq. (16) or (18) with $K_d = 10^4$ ml/g for $Pu(OH)_4$, D_A is predicted to be 10^{-14} to 10^{-15} m²/s which is in close agreement with those 0.5×10^{-14} m²/s determined by experiments in compacted bentonite.

In summary, the free-water model is valid for sandy soil which is dominated by free water. However, for clay soils, clay-species-water interaction need to be considered. This effect has been shown to be important for compacted bentonite. The clay-species water interaction model can be used to generalize the diffusion coefficients by classifying the species into groups according to their chemical behavior. Work is being carried in this regard and also to extend this model to less active clay soils and loose bentonites.

CONCLUSION

A clay-species-water interaction model is developed to describe diffusion in compacted bentonite. The model shows that diffusion depends on the type of diffusing species, the relative size of species and pores, soil density, soil structure, temperature, pore water chemistry and diffusion pathways. The model is shown to agree with experimental results. For some species, the predicted diffusion coefficients are shown to be in excellent agreement with those determined in compacted bentonite.

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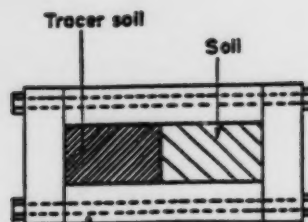
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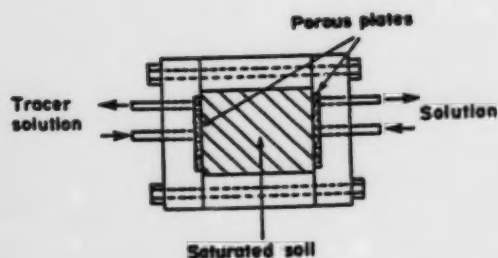
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Table 1. Measured Diffusion Coefficients in compacted bentonite of 1.20 Mg/m³

Species	K _d , ml/g	D _A , m ² /s	D' _A , m ² /s
I ⁻	~0	1 x 10 ⁻¹¹	10 ⁻¹²
Ca ²⁺	20 - 600	1 x 10 ⁻¹¹	10 ⁻¹⁰
Sr ²⁺	1000	3 x 10 ⁻¹²	10 ⁻¹¹



(a)



(b)

Fig.2. Diagram of diffusion cells;

(a) In-diffusion cell

(b) Through-diffusion cell

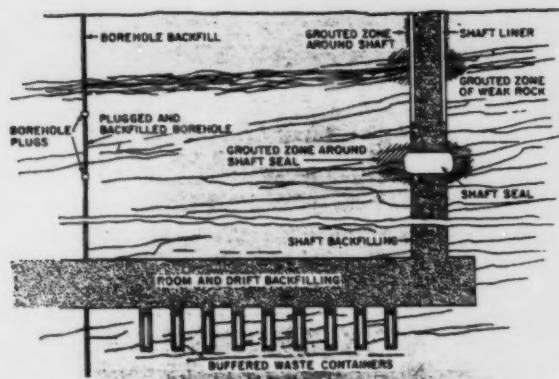


Fig.1. Engineered Barriers and Components of a Nuclear Fuel Waste Disposal Vault

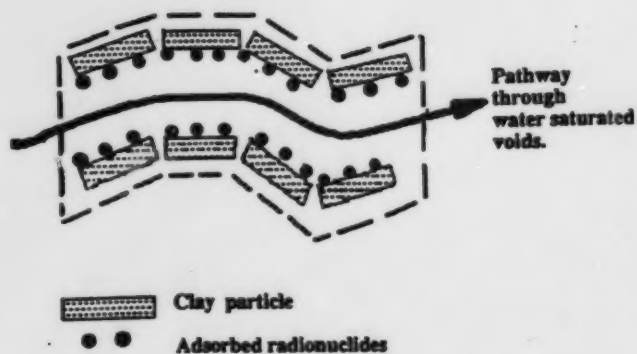


Fig.3. Schematic to illustrate Free-water Model

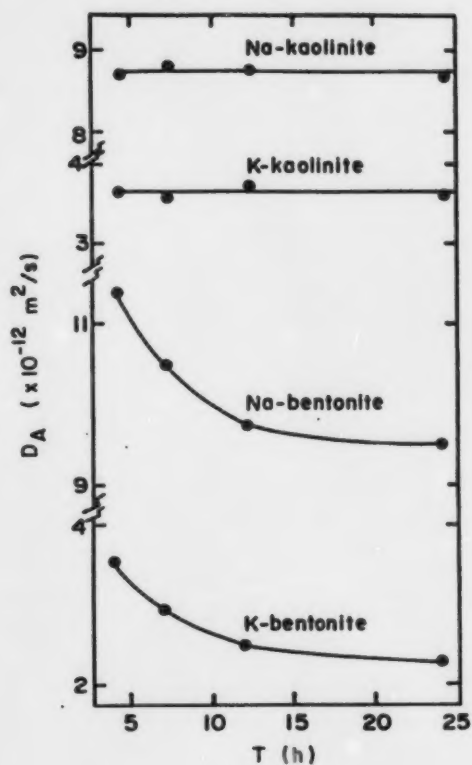


Fig.4. Relationship between apparent diffusion coefficient, D_A , and diffusion time, T , for clays, after Lai(1967).

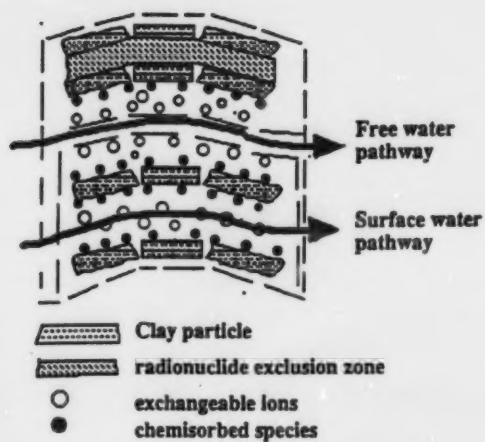


Fig.5. Schematic to illustrate clay-species-water interaction model

ANALYSIS, DEVELOPMENT & TESTING OF A
TENSION CONTROLLED SKYLINE LOG-YARDING CARRIAGE

C. DAYSON

TRIBOLOGY & MECHANICS LABORATORY
INSTITUTE FOR MECHANICAL ENGINEERING
NATIONAL RESEARCH COUNCIL
VANCOUVER, B.C.

ABSTRACT

This paper describes a novel skyline yarding carriage invented and developed at the National Research Council's engineering laboratories in Vancouver.

A skyline system has advantages over other cable systems for long distance haulage of logs and for selective cutting; it has a very tight fixed "skyline" on which the carriage runs. The new NRC carriage has several advantages compared with the standard one; it feeds line to the chokerman who attaches the cables to the logs; it increases the system's capacity for pulling the logs to the skyline and it improves the efficiency of the log's transportation along the skyline. These advantages are achieved with a low cost, all mechanical system which relies on the tensions in the hauling cables to change the mode of operation of the carriage. This patented system is described in detail and a theoretical analysis is presented which allows the performance of the system to be predicted. Two prototypes of the device, which have been built, are described and their performance in field trials is reported.

INTRODUCTION

Today, with the public demand that forestry be carried out in a more sustainable manner, there is within the forest industry a great interest in developing new, more efficient and effective means of selective logging. This is particularly so in British Columbia's coastal rain forests where the demand to preserve much of the remaining old growth forests requires that means be found to increase the yield from second growth forests. Thinning of these forests by selective cutting is one effective way of doing this.

Traditionally these forests were "clear cut" using large cable yarding systems on steep slopes and powerful ground skidders on the flatter terrain. Neither of these systems could be successfully used for selective cutting because of the damage that would be done to the standing trees. Selective logging requires that new methods of extracting the fallen timber be developed using smaller scale and less powerful equipment than was used for clear-cutting. On flatter terrain this may involve ground skidding with small winches or horses and machines such as forwarders which can negotiate narrow trails. On steeper slopes however the only technique which is economically and environmentally appropriate is the use of "skyline" cable systems. A small scale skyline yarding system in which the carriage runs on a taut fixed cable, is illustrated in Figure 1. For selective cutting the skyline system would be set up in cleared corridors spaced at 50 to 100 feet intervals.

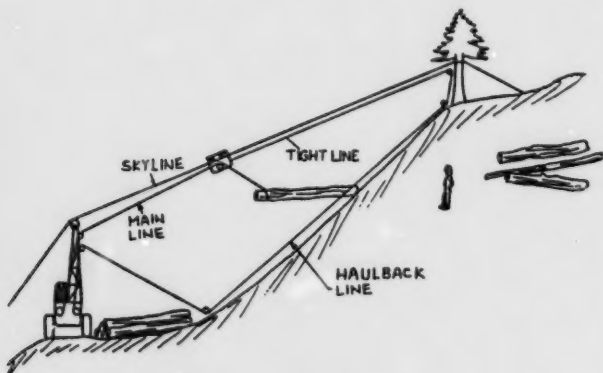


FIGURE 1. A SMALL SCALE SKYLINE YARDING SYSTEM

Experimental selective logging trials with the skyline system have been undertaken a number of times during the last fifteen years and the results of some of these have been documented (1,2). Generally these trials have shown that skyline yarding can be used successfully for thinning on steep or flat terrain. Most of the smaller equipment used in these trials was manufactured in Europe where it has been developed for harvesting the smaller trees that grow there.

The trials also indicated however that this equipment was deficient in certain ways. It was found that the basic skyline equipment suffered from three particular weaknesses. Most importantly it was difficult to "pull slack" mainline out to the logs, particularly in downhill logging situations; see Figure 2.

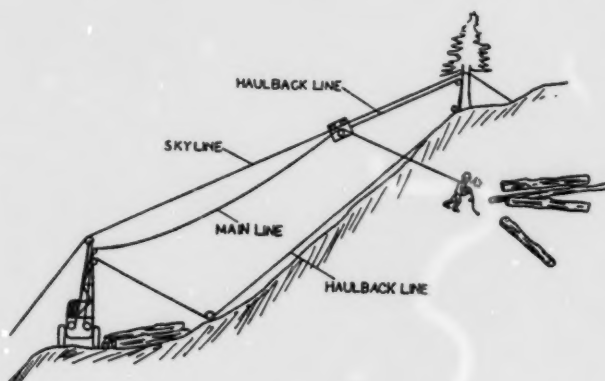


FIGURE 2. "PULLING THE SLACK" IN A DOWN-HILL LOGGING SITUATION

Also during the transportation stage (as shown in Figure 1) it is necessary to maintain a tight haulback tension in order to keep the front end of the log off the ground; this can only be accomplished by braking the haulback line drum on the yarder which results in a waste of energy and causes brake wear. Finally, these low powered systems, typically 50 - 100 HP did not always have enough hauling capacity to "break out" a log from the place where it had been felled. These problems were first brought to the attention of the author by a small logging company on Vancouver Island in 1979. (3)

NOVEL SKYLINE SYSTEM

These difficulties were thoroughly examined and then means were sought to overcome them. After considering a number of approaches an alternative carriage/cable system was conceived (4,5) which would overcome all three of the above disadvantages.

The novel carriage is illustrated in Figure 3 and the new cable arrangement it requires is shown in Figures 4 and 5. It is clear from these drawings that the usual main and haulback lines have been replaced by a single cable which passes through the carriage and whose ends are attached to and controlled by the usual main and haulback winches on the yarder.

Knowing these facts about the carriage design, the way in which it will function to accomplish the performance required can be explained. Thus, by winching in the "haulback" line at the yarder, while braking the outgoing "main" line as shown in Figure 4, a high tension is obtained throughout the cable so that the skyline brake is applied (ie., the carriage is stationary), the dogs open, and "slack" log line is unspooled from the capstan driven

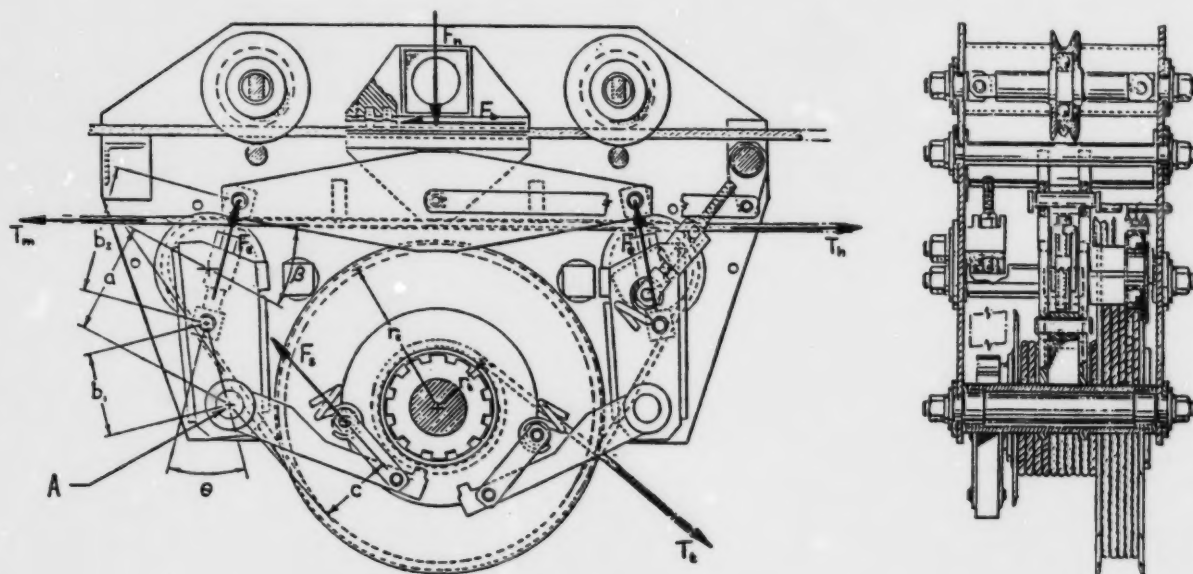


FIGURE 3. THE N.R.C. TENSION CONTROLLED SKYLINE CARRIAGE (FIRST PROTO TYPE)

Figure 3 shows that within the carriage the main/haulback line passes first over a sheave mounted on a spring loaded arm at one end, several times around a capstan, and then over another identically mounted sheave at the other end of the carriage. The capstan is attached to a drum from which the log or "drop" line is unspooled. Also the pivoted arms on which the sheaves are mounted are integral with dogs which unlock or lock the capstan/drum, and a linkage system which respectively applies or releases a skyline brake. The tension springs acting on the dog arms normally hold the dogs in the engaged position, ie. lock the capstan drum and release the skyline brake, when the tension in the hauling line is low on the mainline or the haulback side. Only when the cable tension is high on both sides of the carriage will both dogs disengage so that the capstan/drum is free to turn and the carriage clamps itself (through the brake) to the skyline.

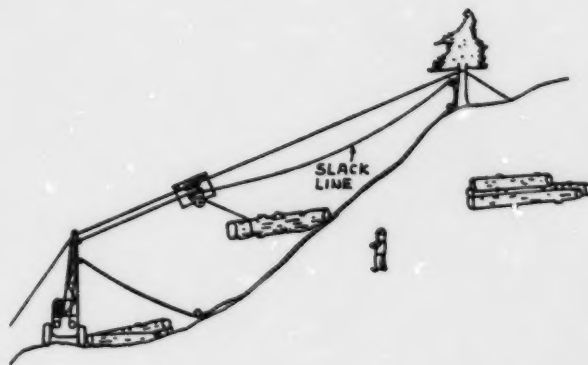


FIGURE 4. THE N.R.C. SKYLINE SYSTEM - PULLING SLACK

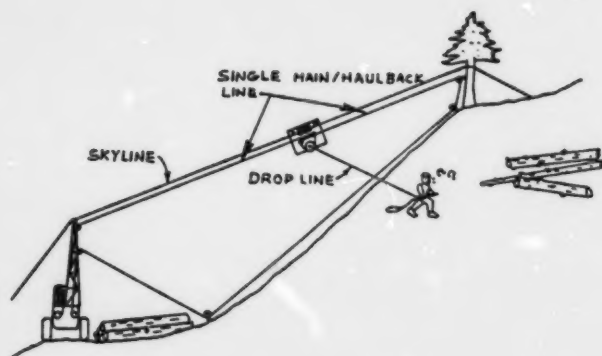


FIGURE 5. THE N.R.C. SKYLINE SYSTEM - TRANSPORTING

drum. The end of this line with the chockers can then be easily carried to the logs.

After the logs have been attached to the drop-line by the chokerman, skidding them between the standing trees and raising them to the skyline is achieved by simply moving the hauling line in the opposite direction, i.e. "reeling in" on the mainline winch, while braking the haulback winch as the line spools out. The required increase in breakout capacity is achieved by having a capstan diameter larger than that of the spooled cable on the drum.

To transport the log along the skyline corridor it is only necessary to release the brake on the haulback winch (i.e. reduce the haulback tension) in order to release the skyline brake and engage the dog on the haulback side (which locks up the log) then the mainline winch hauls the carriage with the log to the landing as shown in Figure 5. The slack haulback line of course ensures efficient transportation and avoids brake wear at the haulback winch. The full sequence of events in transporting a "turn" (a log load) are shown in Figure 6.

FIRST PROTOTYPE

A theoretical model of the carriage was developed in order to assess the feasibility of controlling it by means of the cable tensions. The numerical values used in this analysis apply to the first prototype which closely resembled the description of the carriage described in the patent and which is illustrated in Figure 3. Most of the nomenclature for this analysis is also given in Figure 3. Two limiting operating situations are examined; one where the skyline brake is most likely to slip; and the other where the slip is most likely to occur between the main/haulback line and the capstan.

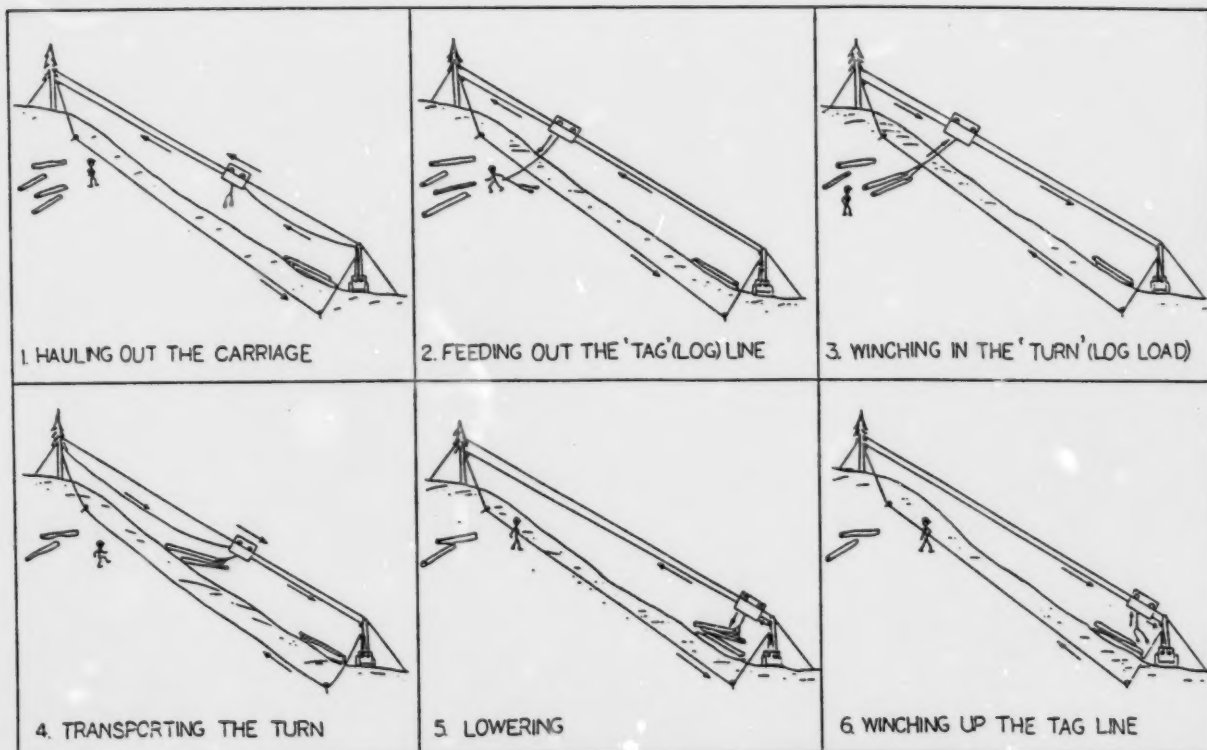


FIGURE 6. SEQUENCES IN THE OPERATION OF THE TENSION CONTROLLED CARRIAGE

Case One: Skyline Brake Slip Limit

This limit of operation would most easily occur in a steep downhill logging situation where a log is being raised vertically to the carriage, as shown in Figure 7. Where there is insufficient skyline brake capacity, the carriage would move downhill under the action of the log weight and the mainline tension. For no slip, therefore:

$$T_h + F_b > T_m + T_t \sin \phi \quad (1)$$

Where μ_b is the limiting static friction coefficient for the skyline/brake block combination, it follows from Figure 3 that

$$F_b = 2\mu_b F_n \quad (2)$$

and

$$F_n = 2F_s \quad (3)$$

Taking moments about A for the equilibrium of the haulback line pivot arm/dog:

$$F_s b_1 \sin \theta = 2T_h a \cos \beta - CF_s \quad (4)$$

From Equations 2, 3 and 4

$$F_b = \frac{4\mu_b (2aT_h \cos \beta - CF_s)}{b_1 \sin \theta} \quad (5)$$

Also, for equilibrium of the drum/capstan when the dog clutch is disengaged:

$$(T_m - T_h) r_c = T_t r_t \quad (6)$$

$$T_m = \frac{T_t r_t}{r_c} + T_h \quad (7)$$

Substituting for F_b and T_m , therefore, in Equation 1, the lowest value of T_h to prevent brake slip is given by:

$$T_h \geq \frac{b_1 \sin \theta}{8\mu_b a \cos \beta} \left(\frac{4\mu_b CF_s}{b_1 \sin \theta} + \frac{T_t r_t}{r_c} + T_t \sin \phi \right) \quad (8)$$

Where the critical dimensions are given by:

$$\begin{aligned} a &= 9.25 \text{ in (0.235 m)} \\ r_c &= 11 \text{ in (0.2 m)} \quad \phi = 32.5^\circ \\ b_1 &= 6.0 \text{ in (0.152 m)} \\ r_t &= 5.5 \text{ in (0.14 m)} \quad \theta = 30^\circ \\ b_2 &= 9.0 \text{ in (0.229 m)} \end{aligned}$$

and the maximum design load, the spring moment, and μ_b are given by:

$$\begin{aligned} T_t &= 7,500 \text{ lb (3.36 x 10}^4 \text{ Newtons)} \\ CF_s &= 10,000 \text{ in lb (1,130 metre Newtons)} \\ \mu_b &= 0.15 \end{aligned}$$

then on a 100 percent slope ($\phi = 45^\circ$), the minimum haulback tension to prevent slip of the skyline brake is given by Equation 8:

$$T_h \geq 3,540 \text{ lbs (15.7 x 10}^4 \text{ Newtons)}$$

The corresponding mainline tension is by Equation 7:

$$T_m = 7,290 \text{ lb (3.24 x 10}^4 \text{ Newtons)}$$

Thus although a high haulback tension must be maintained it is clear that even under this adverse condition the log can be raised to the skyline.

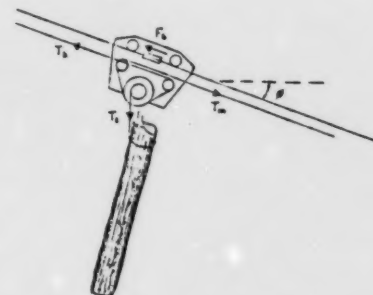


FIGURE 7. POTENTIAL SKYLINE BRAKE SLIP CONDITION

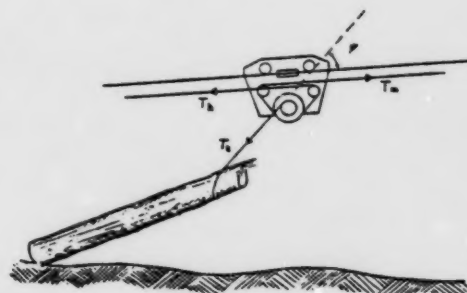


FIGURE 8. POTENTIAL CAPSTAN/CABLE SLIP CONDITION

Case Two: Capstan Slip Limit

This limiting operational situation occurs

when the log is being transported and the skyline is close to the ground so that the angle between the tagline and skyline is at its lowest, as shown in Figure 8. For longitudinal equilibrium,

$$T_a - T_h = T_c \cos \psi \quad (9)$$

Where the haulback line dog is to remain engaged (to ensure the log is locked up) during transportation, then T_h must be given by:

$$2T_h \alpha \cos \beta < CF_s \quad (10)$$

Also where μ_c is the limiting static friction coefficient for the cable on the capstan and α is the angle of wrap the maximum ratio of main to haulback line tensions is given by:

$$\left(\frac{T_a}{T_h}\right)_m = e^{\mu_c \alpha} \quad (11)$$

Taking the extreme case of the angle $\psi = 30^\circ$, Equation 9 gives:

$$T_a - T_h = 7500 \cos 30^\circ = 6.495 \text{ lbs.} \\ (2.89 \times 10^4 \text{ Newtons})$$

However, the maximum value of T_h is given according to Equation 10, i.e.,

$$T_h < \frac{CF_s}{2\alpha \cos \beta}$$

Now, where $\mu_c = 0.15$ (for marginally oil lubricated surfaces)

$$\alpha = \frac{20\pi}{3} \text{ radians } (3 \frac{1}{3} \text{ turns})$$

Equation 11 gives

$$\frac{T_a}{T_h} = e^{\mu_c \alpha} = 23.14$$

Therefore

$$\frac{T_a}{T_h} - 1 = \frac{T_a - T_h}{T_h} = 22.14$$

and

$$T_h = \frac{T_a - T_h}{22.14} = 293 \text{ lbs } (0.131 \times 10^4 \text{ Newtons})$$

Thus even where a low capstan friction coefficient is assumed, the haulback tension required to prevent slip on the capstan (293 lbs) is well below the maximum allowed (641 lbs).

Although the idea of this skyline system was conceived in 1980, the design and development of a prototype carriage was not initiated until 1984. The first prototype was designed to carry a log load of 7500 lbs. and the drop line was 130 feet long. Also it had a 2:1 ratio of capstan diameter to that of the drop-line drum for increasing the carriage's breakout capacity. The analysis and numerical calculations above indicate that such a carriage would be capable of being effectively controlled by means of the cable tensions even under the adverse operating conditions considered there. The carriage that was designed and built on the basis of this design data is shown in Figure 9. Its construction was similar to that described in the patent (see Figure 3). It weighed 900 lbs.

This prototype was tested in the woods several times between 1985 and 1987 in co-operation with the yarder manufacturer Skylead Ltd. of Enderby, B.C.; see Figure 10. These tests proved that the new skyline system was effective and practical. The new system was capable of pulling slack line and it provided an enhanced log breakout capability (6). However the prototype carriage as constructed was found to have several deficiencies. It was heavy because a high safety factor i.e. 2, was used in the design. Later it was realised that such a high factor was not needed because the carriage is not subjected to high impacts due to of the cushioning effect of the cables. Another more serious problem however was the lack of a "fail-safe" capability during the log chocking and un-chocking stages of the yarding operation.

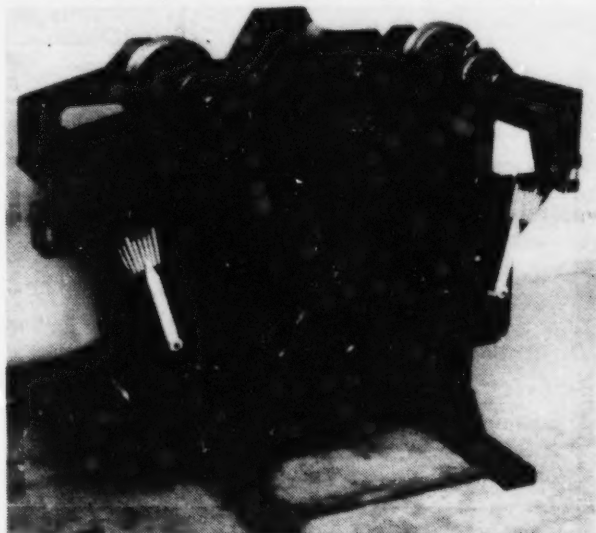


FIGURE 9. FIRST PROTOTYPE CARRIAGE



FIGURE 10. TRANSPORTING A "TURN" WITH THE FIRST PROTOTYPE

If, for instance, the mainline was inadvertently released while these operations were occurring on an uphill logging operation, the skyline brake would release, and the carriage immediately run away, creating a highly hazardous situation. Thirdly, this prototype was not capable of passing an intermediate skyline support. This is an important requirement if the system is to function effectively in a wide range of terrain.

In view of these three disadvantages it was decided after three field trials that a second prototype was required which would be designed so that it would not be subject to any of these deficiencies. (6).

SECOND PROTOTYPE

In 1988 a second prototype carriage was designed. In order to reduce its size and weight compared to the first one, the load capacity and the safety factor were considerably lower ie. 4,000 lbs and 1.3 respectively. In order to solve the safety problem the skyline brake was completely redesigned. Instead of being actuated and powered by a linkage connected to the pivoting pulley arms, a locking brake was used which would not release in the event that a hauling line was unintentionally let go. Finally in order that the carriage would be able to pass an intermediate skyline support it was designed asymmetrical with a single reinforced frame plate. This second prototype which weighed only about 400 lbs is shown in Figures 11 and 12. The first of these shows the capstan/drum side with its two sheaves mounted on pivoted arms and the capstan and drop line drum. This is also the side on which the carriage passes the intermediate skyline support. Figure 12 shows the other side of the carriage with the dogs (connected to the sheave pivot arms on the other side), the dog wheel which is integral with the drop-line drum and capstan on the other side, and the hydraulic master cylinders and lines which provide the signals to trigger the skyline locking brake.

Although the skyline brake is triggered hydraulically, the high braking force is obtained by the tapered brake blocks wedging against rollers; see Figure 13. Clearly the brake acts in either direction so that the carriage is prevented from moving away from the yarder (when pulling slack), or towards the yarder (when skidding the log to the skyline). It should be noted however that once applied, this brake can only be released by "unlocking" it, which involves moving the carriage slightly in the direction opposite to that which the brake was preventing motion.

For example, after skidding and raising the log when the carriage is prevented from moving towards the yarder by the brake, it has to be run a few inches or a foot away from the yarder to release the brake before the carriage can be moved towards the yarder with its log load.



FIGURE 11. SECOND PROTOTYPE - CAPSTAN/DRUM SIDE



FIGURE 12. SECOND PROTOTYPE - DOG CLUTCH SIDE

The analysis that was performed in order to properly design the locking skyline brake is of interest. Figure 13 illustrates this brake and the relatively complex geometric and frictional aspects of the situation. Clearly in order to lock the steel rope between the hardened steel brake blocks the friction between the rope and the blocks must be sufficient to cause sliding between the rollers and their shaft. The resultant force between the roller and its shaft must therefore act at the limiting frictional angle σ_r which is given by:

$$\sigma_r = \tan^{-1} \mu_r \quad (12)$$

where μ_r is the static friction coefficient for the roller bearing/shaft contact. There are only two forces acting on the roller; so for it to be in equilibrium, the above force and the force between it and the tapered brake block must be of the same magnitude and act in the same line AB. Also it follows that for equilibrium of the brake block which is also only subjected to two forces, the resultant force between the block and the rope must again be of the same magnitude and act in the same direction, ie. all the forces are equal and act in the same line. It also follows therefore from Figure 13 that

$$\frac{R}{\sin(\pi - \sigma_r)} = \frac{r}{\sin \sigma_c} \quad (13)$$

Thence from Equation 12 and 13

$$\sigma_c' = \sin^{-1} \left(\frac{r}{R} \sin(\pi - \tan^{-1} \mu_r) \right) \quad (14)$$

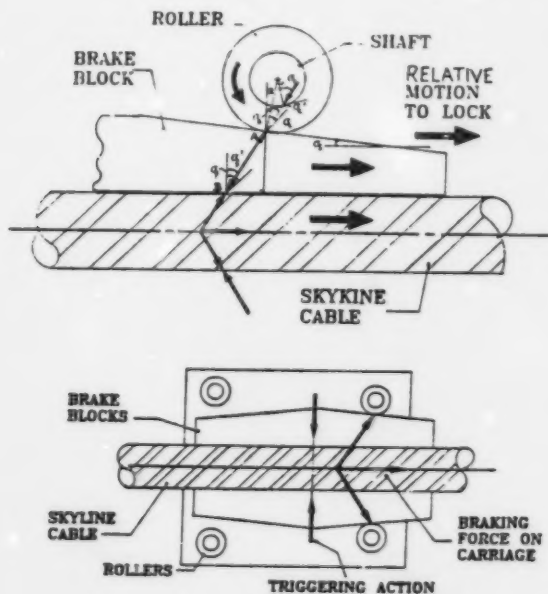


FIGURE 13. THE LOCKING SKYLINE BRAKE

For there to be sufficient traction between the roller and the block to cause sliding between the roller and the shaft it follows that

$$\sigma_c = \tan^{-1} \mu_c > \sigma_c' = \tan^{-1} \mu_c' \quad (15)$$

Also and more critically the friction between the brake blocks and the steel cable must be great enough if the brake is to lock effectively. Since it follows from Figure 13 that

$$\sigma_c' = \sigma_c + \sigma_s \quad (16)$$

and from Equation 14 and 16 that

$$\sigma_c' = \sigma_s + \sin^{-1} \left(\frac{r}{R} \sin(\pi - \tan^{-1} \mu_r) \right) \quad (17)$$

then the lowest possible friction coefficient between the rope and the block at which the brake will function properly is given by μ_c where

$$\sigma_c = \tan^{-1} \mu_c > \sigma_c' = \tan^{-1} \mu_c' \quad (18)$$

Assuming that for grease lubrication

$$\mu_r = 0.1$$

And for $R = 0.625$ in. and $r = 0.375$ in. it follows from Equation 14 and 15 that

$$\sigma_c' = 3.42^\circ, \mu_c > 0.06$$

Since μ_c for oily surfaces is typically about 0.1 there is clearly adequate traction between the roller and the block. For $\sigma_c = 4^\circ$ it follows from Equations 17 and 18 that

$$\sigma_c' = 7.42^\circ, \mu_c > 0.13$$

Since the friction μ_c between a steel rope and steel blocks will be typically 0.2, such a brake will therefore function effectively.

Due to a lack of resources the development process for the second prototype has been slow. However since 1989, when it was first tested in the woods, four trials have been undertaken. The first demonstrated clearly that the disadvantages associated with the first carriage had been successfully overcome. Because of its low weight it was much easier to handle, and with the locking skyline brake it was shown that there was no "runaway" hazard. Also its capacity was higher than anticipated with logs weighing an estimated 6 - 7,000 lbs being hauled uphill to a landing. However, for more than a year progress was held up by intermittent problems with the mechanical/hydraulic triggering system for the skyline brake.

In early 1991 however it was realised that this unnecessarily complicated system could be replaced by a much better all hydraulic system. At the same time as this modification was made, the dog retention spring system was also modified in order to simplify it and improve its performance.

In the last field trial to date (at Port McNeil in June 1991) these modifications were found to have been very effective.

The whole skyline system functioned flawlessly over a two day test period during which time a truckload of thinned timber was yarded. Also during this trial a new operator technique was developed where the cables were controlled using just the winch clutches only, rather than the clutches and the brakes. This meant that most of the time the operator was only using two levers instead of four. The brakes were only set when the carriage was to be held stationary for a period of time, e.g. while the logs are being choked.

The performance of the system was clearly advantageous compared with the standard skyline. The testing was done in an uphill logging situation where the terrain was less steep at the lower end of the skyline; see Figure 14. Slack line from the carriage drum was simply reeled out and the chockers easily carried out to the logs. On the other hand with the standard carriage the chockerman required the assistance of the faller to haul slack mainline. Also during the transportation phase the energy used was lower; the logs were moved up the skyline to the landing at a noticeably lower power level or alternatively they were moved at a higher speed with a resulting increase in productivity.



FIGURE 14. YARDING WITH THE SECOND PROTO TYPE

However, another minor problem was discovered at the conclusion of the two day trial. The section of the main/haulback line that runs through the carriage when it is pulling slack or skidding the logs to the skyline, was becoming worn and strands were breaking. A subsequent investigation of this problem which was first thought to be due to the micro-sliding action of the cable on the capstan is now thought to have been caused by excessive cable scrubbing and flexing at the small sheaves. These are now being replaced by larger diameter sheaves.

CONCLUSION

A further and final field test is to be undertaken during March 1992. Thereafter the project is to be passed to the licensed manufacturer¹ who intends to build a production prototype prior to manufacturing the carriage for the expanding market now developing for selective logging equipment.

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¹ Ramsay Machine Works Ltd., Victoria, B.C.

FREE VIBRATION OF RECTANGULAR PLATES
WITH NON-UNIFORM ROTATIONAL ELASTIC
EDGE RESTRAINT

by

D.J. Gorman, Professor
Department of Mechanical Engineering
University of Ottawa
770 King Edward Ave.
Ottawa, K1N 6N5 CANADA

ABSTRACT

A general analytical solution based on the superposition method is developed for conducting a free vibration analysis of rectangular plates with variable elastic rotational support along the edges. Lateral motion of the plate edges is forbidden. Verification tests are conducted to demonstrate that correct frequency limits are approached as the stiffness distributions approach their limits of zero and infinity. There is no restriction on the manner in which the rotational stiffnesses may vary along the plate edges.

INTRODUCTION

There are numerous papers to be found in the literature which treat the problem of free vibration analysis of thin rectangular plates with uniform elastic support along the edges. The term "uniform" implies that the elastic stiffness is constant, or uniform, along each edge.

Among the more simple problems in this area, are those where one pair of opposite edges of the plate have simple support, with elastic support acting on one or both of the remaining edges. An exact Lévy-type solution is easily achieved for such plates. In the more general problem, where a pair of opposite simply supported edges does not exist, obtaining accurate solutions is a much more difficult undertaking.

In recent years the author has presented solutions to this more general problem. These solutions were achieved by exploiting the method of superposition and they addressed the problem where the elastic stiffness distributions are symmetric with respect to the plate central axis [1], and the even more general problem where no symmetry in the elastic stiffness distributions is required [2].

In this paper the superposition technique is extended to handle the problem

of plates with rotational elastic stiffness distributed along the edges, where no restriction whatever is imposed on the stiffness distributions along any edge. It will be seen that accurate and rapidly converging solutions are obtained for this family of problems.

MATHEMATICAL PROCEDURE

The procedure followed here is primarily an extension of the method as employed in an earlier paper [2]. The reader should study the earlier work as many details presented there will not be repeated here.

Let us begin by considering the rectangular plate problem as depicted in Figure 1. Edges 2,3, and 4, as denoted in the figure, have simple support. The remaining edge, edge No. 1, differs from the others in that there is a distributed linear elastic resistance to rotation along this edge.

With the plate in vibratory motion, it is known that the following relationship must be satisfied along the elastically supported edge,

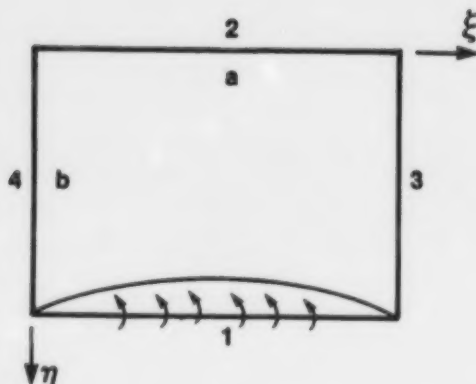


FIGURE 1. Rectangular plate with simple support on three edges and distributed rotational stiffness along the fourth edge where lateral motion is forbidden. Note numbers used to designate edges.

$$\frac{M(\xi)b^2}{aD} = K_1(\xi) \left. \frac{\partial w(\xi, \eta)}{\partial \eta} \right|_{\eta=1} \quad (1)$$

where

$$K_1(\xi) = \frac{k(\xi)b}{D} \quad (2)$$

and $M(\xi)b^2/aD$ is a distributed harmonic bending moment acting along this latter edge. The solution for the shape of the plate vibration is taken in the form proposed by Lévy as,

$$w(\xi, \eta) = \sum_{m=1,2}^{\infty} Y_m(\eta) \sin m\pi\xi \quad (3)$$

A solution for the function, $Y_m(\eta)$, is obtained by substituting equation (3) into the governing differential equation. Two constants appearing in these latter solutions are immediately eliminated through enforcement of the boundary conditions along the edge, $\eta=0$. We thus obtain,

for $\lambda^2 > (m\pi)^2$

$$Y_m(\eta) = A_m \sinh \beta_m \eta + B_m \sin \gamma_m \quad (4)$$

and, for $\lambda^2 < (m\pi)^2$,

$$Y_m(\eta) = A_m \sinh \beta_m \eta + B_m \sinh \gamma_m \quad (5)$$

where, A_m and B_m are constants to be determined

$$\beta_m^2 = \phi^2(\lambda^2 + (m\pi)^2), \text{ and } \gamma_m^2$$

$$= \phi^2(\lambda^2 - (m\pi)^2), \text{ or}$$

$$\phi^2((m\pi)^2 - \lambda^2), \text{ whichever is real.}$$

Enforcement of the condition of zero lateral displacement along the edge, $\eta=1$, permits us to write,

$$Y_m(\eta) = A_m(\sinh \beta_m \eta + \theta_{1m} \sin \gamma_m \eta) \quad (6)$$

and

$$Y_m(\eta) = A_m(\sinh \beta_m \eta + \theta_{2m} \sinh \gamma_m \eta) \quad (7)$$

where,

$$\theta_{1m} = \frac{-\sinh \beta_m}{\sin \gamma_m}, \text{ and } \theta_{2m} = \frac{-\sinh \beta_m}{\sinh \gamma_m}$$

Representing the amplitude of the distributed harmonic bending moment acting along the edge, $\eta=1$, as,

$$\frac{Mb^2}{aD} \Big|_{\eta=1} = \sum_{m=1,2}^{\infty} E_m \sin m\pi\xi \quad (8)$$

and enforcing dynamic moment equilibrium along this edge we obtain,

for $\lambda^2 > (m\pi)^2$

$$Y_m(\eta) = E_m(\theta_{11m} \sinh \beta_m \eta + \theta_{13m} \sin \gamma_m \eta) \quad (9)$$

and, for $\lambda^2 < (m\pi)^2$

$$Y_m(\eta) = E_m(\theta_{22m} \sinh \beta_m \eta + \theta_{23m} \sinh \gamma_m \eta) \quad (10)$$

where

$$\theta_{11m} = \frac{-1}{(\beta_m^2 + \gamma_m^2) \sinh \beta_m};$$

$$\theta_{13m} = \frac{-1}{(\beta_m^2 + \gamma_m^2) \sin \gamma_m}$$

$$\theta_{22m} = \frac{-1}{(\beta_m^2 - \gamma_m^2) \sinh \beta_m};$$

$$\text{and } \theta_{23m} = \frac{1}{(\beta_m^2 - \gamma_m^2) \sinh \gamma_m}$$

Based on equations 1, 3, 9 and 10, we may now write

$$\begin{aligned} \frac{M(\xi)b^2}{aD} \Big|_{\eta=1} = K_1(\xi) \left\{ \sum_{m=1}^{\infty} E_m [\theta_{11m} \beta_m \cosh \beta_m \right. \\ \left. \sin m\pi\xi + \sum_{m=k+1}^{\infty} E_m [\theta_{22m} \beta_m \cosh \beta_m \right. \\ \left. + \theta_{23m} \gamma_m \cosh \gamma_m] \sin m\pi\xi \right\} \end{aligned} \quad (11)$$

where the first summation includes only terms for which $\lambda^2 > (m\pi)^2$.

Introducing the symbol H_m where, for $\lambda^2 > (m\pi)^2$

$$H_m = \theta_{11m} \beta_m \cosh \beta_m + \theta_{13m} \gamma_m \cos \gamma_m \quad (12)$$

and, for $\lambda^2 < (m\pi)^2$

$$H_m = \theta_{22m} \beta_m \cosh \beta_m + \theta_{23m} \gamma_m \cosh \gamma_m \quad (13)$$

Equation (11) is written as,

$$\frac{M(\xi)b^2}{aD} \Big|_{\eta=1} = K_1(\xi) \sum_{m=1,2}^{\infty} E_m H_m \sin m\pi\xi \quad (14)$$

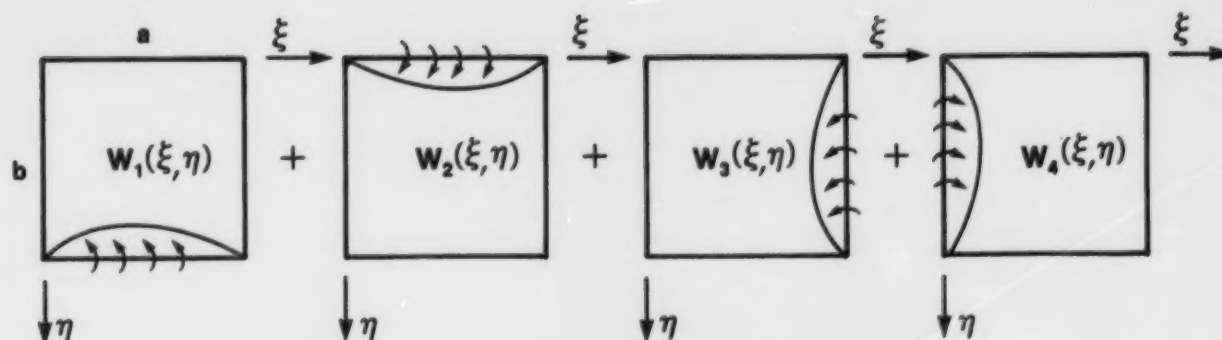


FIGURE 2. Four building blocks with rotational stiffness distributed along edges, 1,2,3, and 4, respectively. These building blocks are superimposed to obtain general solution.

The left hand side of equation (14) is available in a sine series (Eq. 8), and it will be obvious that the right hand side can be expanded in the same series.

We express the stiffness distribution $K(\xi)$, as,

$$K(\xi) = \sum_{i=1}^{\infty} ET_1(i) \sin i\xi \quad (15)$$

Consider now the n^{th} term in the Fourier sine series expansion of each side of equation (14). We obtain,

$$E_n = 2 \sum_{m=1}^{\infty} E_m H_m \int_0^1 \sum_{i=1}^{\infty} ET_1(i) \sin i\xi \sin m\xi \sin n\xi d\xi \quad (16)$$

Taking K as the upper limit of the series or equation (3), and hence equation (8), the parameter n of equation (16) can take on K distinct integer values. The integrals of equation (16) are easily called by a simple computer algorithm.

Corresponding to each value of n , equation (16) provides us with an algebraic equation relating the K unknown Fourier coefficients. We, therefore, obtain K such equations. If our interest lay in solving the problem of a rectangular plate with three simply supported edges and one edge with lateral displacement forbidden and subjected to rotational elastic support of variable stiffness we would have our solution in equation (16). We would need only to set up the eigenvalue matrix based on the coefficient matrix of the K simultaneous homogenous algebraic equations discussed above. We would then search for the eigenvalues, λ^2 , which cause the determinant of this eigenvalue matrix to vanish.

Since we are interested in the general case where elastic rotational stiffness may be distributed along all four edges, we now turn to the four building block solutions depicted in Figure 2. The solution for the first building block is identical to the solution already discussed. Solution for the second building block is extracted from the first by replacing the variable η with the

quantity $(1-\eta)$. A solution for the third building block is also extracted from the first through interchange of the variables ξ and η , and replacing the plate aspect ratio ϕ with its inverse. The fourth building block solution is extracted from the third through replacing the variable ξ with $(1-\xi)$.

We now obtain the required general solution by superimposing these four building blocks, one upon the other.

Let us consider conditions along the edge, $\eta=1$, of such an assemblage of solutions. A review of the mathematical procedure indicates that equation (16), as formulated earlier, is still applicable except that we must now enter all four building block contributions toward displacement along the edge of interest into the equation. It is seen that there will, therefore, be three extra summations on the right hand side of the equation. Some of the required integrals (pertaining to the third and fourth building blocks) will differ slightly from those of equation (16) and will involve hyperbolic functions. This presents no difficulty.

We find that pertaining to edge, $\eta=1$, we now have K equations relating the $4K$ unknown Fourier coefficients associated with the displacement solutions for the four building blocks. It will be obvious, however, that we can write K equations for each of the other three edges. We then have $4K$ homogenous algebraic equations upon which our general eigenvalue matrix is based. Eigenvalues are determined by searching for those values of λ^2 which cause the determinant to vanish, as discussed earlier. Mode shapes are determined by setting one of the non-zero displacement Fourier coefficients equal to unity and solving the resultant set of non-homogenous equations for the remaining coefficients. A schematic representation of the general eigenvalue matrix, based on three-term representations of the building block displacements is presented in Figure 3.

PRESENTATION OF RESULTS

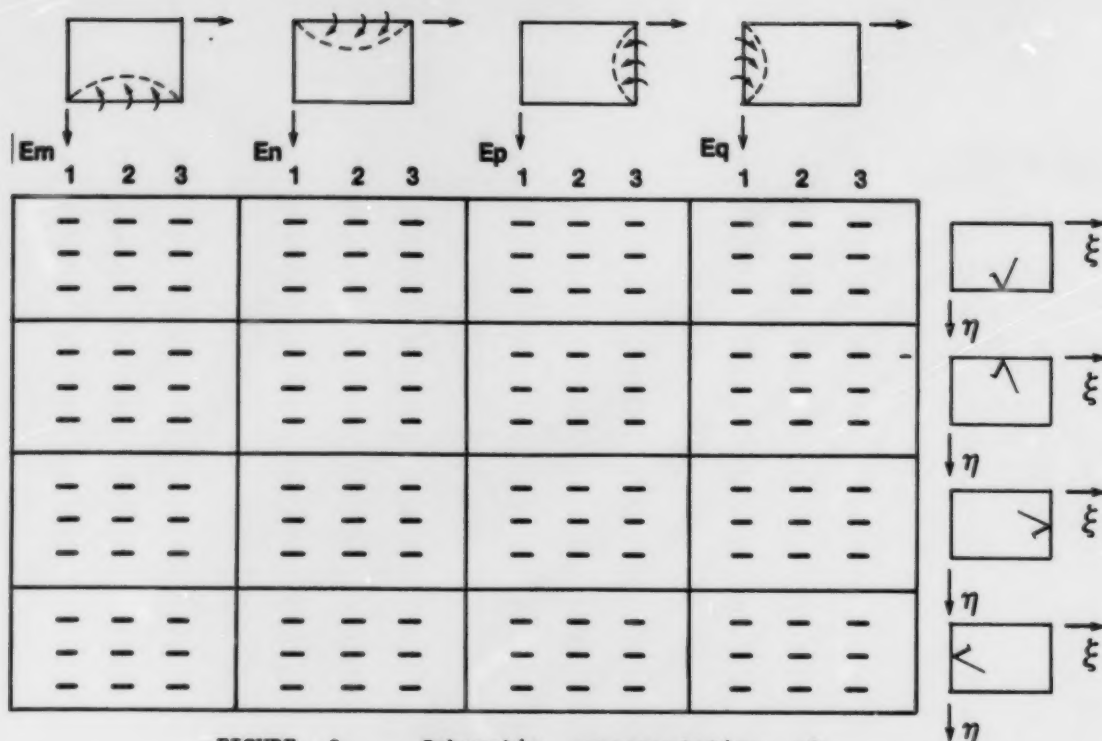


FIGURE 3. Schematic representation of eigenvalue matrix for general problem based on three term expansions of solution series. Inserts at right indicate edges along which moment-slope equilibrium is enforced.

It will be appreciated that it is not possible to present a wide range of results which would cover all possible designers' needs related to this problem. Rather, it is deemed prudent to try and achieve two goals. One is to plot results which will help establish that the computed eigenvalues approach the known exact limits as the support stiffnesses approach their natural limits of zero or infinity. Such exercises serve as verification tests for the theory and the computational procedure. The second goal is to provide tabulated digital results against which other researchers will be able to compare their findings.

A plot of the first mode eigenvalue vs rotational stiffness amplitude "A" for a square plate with elastic rotational support along one edge only is presented in Figure 4. The remaining three edges have simple support. The stiffness distribution here is represented by a single half sine wave of amplitude "A". Analysis is obtained by utilizing the upper left segment only of the matrix as depicted in Figure 3. It is seen that the eigenvalue of Figure 4 approaches the correct limits of zero and 23.65 as amplitude "A" approaches zero and infinity, respectively. The latter limit is, of course, that of a square plate with simple support on three edges and a clamped condition along the other.

We consider next the first mode free vibration of a square plate with uniform

rotational stiffness of amplitude "A" distributed along all four edges (Figure 5). Lower and upper limits for the eigenvalue will evidently be those of a plate with simple support, on all edges and a fully clamped plate. Again, the correct eigenvalue limits are seen to be approached.

We turn next to the tabulation of some computed results. Leissa et al have published some results for a square plate with two opposite edges simply supported and rotational stiffness with a parabolic distribution acting on the other two edges [3]. The basic parabolic distribution is expressed as $\xi(1-\xi)$, the function being symmetric about the mid-point of the edge. The stiffness coefficient K_0 , as defined by Leissa et al is utilized in making comparisons. Results are compared in Table 1. Only the upper left quadrant of the matrix of Figure 3 was utilized in the present analysis. In the limiting cases in the table no computation is necessary. It will be noted that results of the present analysis are slightly higher. Two summation limits are necessary in the present analysis. One relates to the number of terms utilized in the displacement series and the other relates to the number of Fourier terms required to represent the parabolic stiffness distribution. Values of 15 and 9 were used here, respectively. Very good convergence was encountered.

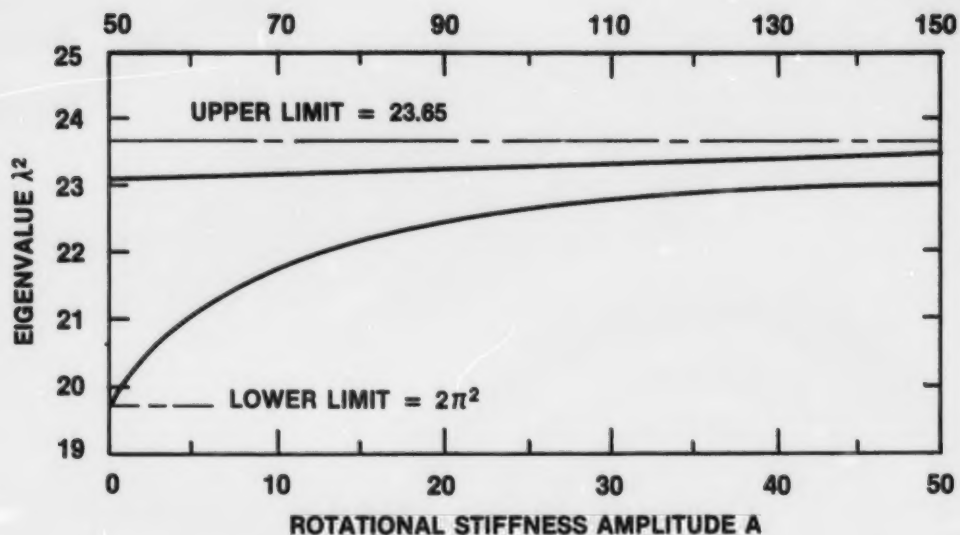


FIGURE 4. First mode eigenvalue vs. amplitude A of single half sine wave representation of rotational stiffness on edge 1 of square plate.

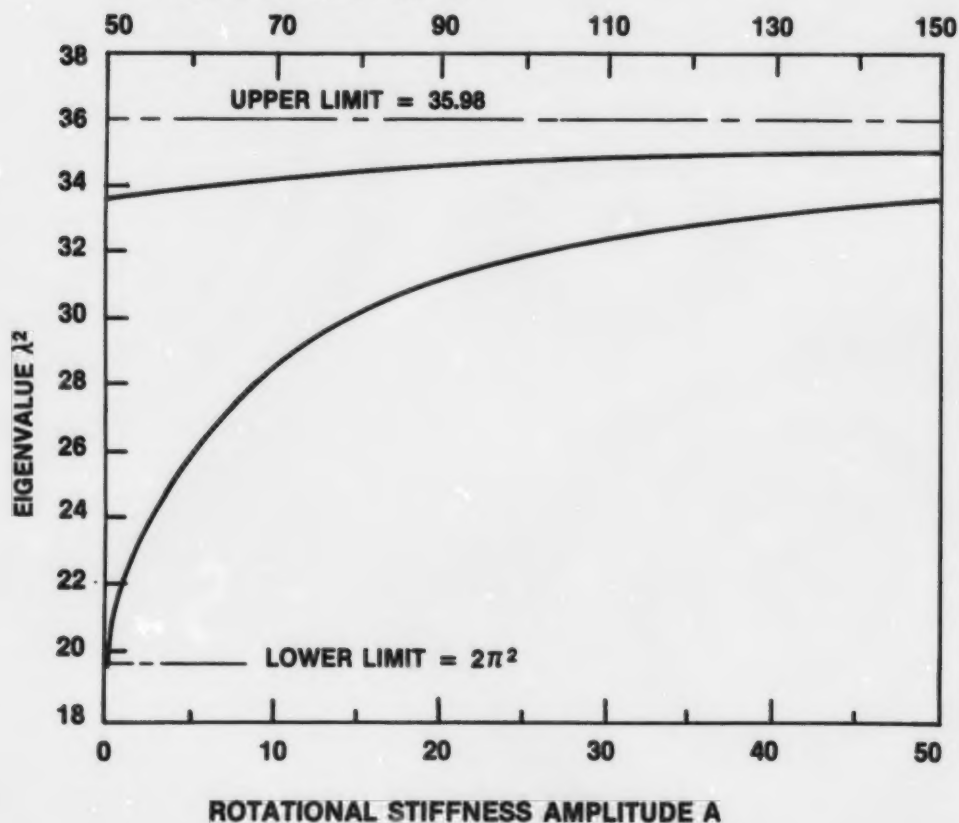


FIGURE 5. First mode eigenvalue vs. amplitude A of uniformly distributed rotational stiffness on all four edges of square plate.

TABLE 1

Computed First Mode Eigenvalues for a Square Plate with Equal Parabolically Distributed Rotational Stiffnesses Along Two Opposite Edges.

$K_0 a^3/D$	A*	B*	(Present)
0	$2\pi^2$	-	-
1	19.915	20.114	19.95
10	21.235	24.068	21.48
100	25.799	28.193	26.15
∞	28.951	-	-

* A and B represent the "more accurate" and Rayleigh-Ritz solutions, respectively, of Leissa et al.

Finally, in Table 2, first mode eigenvalues are tabulated for a square plate with equal parabolically distributed rotational stiffnesses along all edges. Distribution along the edge, $\eta=1$, is given by $K_1 \xi(1-\xi)$. Eigenvalues are, therefore, tabulated against the parameter K_1 . This parameter is incremented in such a way as to give approximately equal intervals in the tabulated eigenvalues.

$ET_1(i)$	i th coefficient in Fourier series representing distributed rotational stiffness along edge 1 of plate
k	total number of terms used in series representation of plate lateral displacement
k^*	number of terms in first summation of series representing plate lateral displacement
$k(\xi)$	rotational stiffness along edge of plate
$K_1(\xi)$	dimensionless rotational stiffness distributed along plate edge 1 = $k(\xi) b/D$
M	bending moment
Mb^2/aD	dimensionless bending moment
W	plate lateral displacement divided by edge length a
1,2,3,4	subscripts designating plate edge (see Fig. 1)
λ^2	free vibration eigenvalue = $\omega^2 \sqrt{\rho/D}$
ω	circular frequency of plate free vibration
ρ	mass per unit area of plate
ϕ	plate aspect ratio = b/a
ξ	distance along plate coordinate direction divided by a
η	distance along plate coordinate direction divided by b

TABLE 2

Computed First Mode Eigenvalues for Square Plate with Equal Parabolically Distributed Stiffnesses Along All Edges.

K_1	0	2.5	5	10	15	20
λ^2	19.74	20.75	21.64	23.12	24.32	25.31
K_1	30	40	50	60	80	100
λ^2	26.86	28.01	28.90	29.61	30.68	31.44
K_1	140	180	260	340	420	500
λ^2	32.45	33.09	33.87	34.31	34.60	34.8

DISCUSSIONS AND CONCLUSIONS

An analytical type solution is obtained for the free vibration of rectangular plates with rotary elastic stiffness of any desired distribution along the edges. Any sub-set of this set of problems can be resolved by deleting an appropriate portion of the matrix; for example, the same plate with rotary stiffness along three edges, only. Solution is obtained by the superposition of a judiciously selected set of building blocks. Convergence is found to be rapid and eigenvalues accurate to four significant digits are easily obtained. To the author's knowledge, this represents the first comprehensive solution to this practical industrial problem.

LIST OF SYMBOLS

a, b	edge lengths of rectangular plate
D	plate flexural rigidity

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ELECTRICALLY-INDUCED STRESSES AND DEFLECTION IN MULTIPLE PLATES

Jih-Perng Hu and Paul R. Tichler
Brookhaven National Laboratory
Upton, NY 11973
U.S.A.

ABSTRACT

Thermohydraulic tests are being planned at the High Flux Beam Reactor of Brookhaven National Laboratory, in which direct electrical heating of metal plates will simulate decay heating in parallel plate-type fuel elements. The required currents are high if plates are made of metal with a low electrical resistance, such as aluminum. These high currents will induce either attractive or repulsive forces between adjacent current-carrying plates. Such forces, if strong enough, will cause the plates to deflect and so change the geometry of the coolant channel between the plates. Since this is undesirable, an analysis has been made to evaluate the magnitude of the deflection and related stresses.

In contrast to earlier publications in which either a concentrated or a uniform load was assumed, in this paper an exact force distribution on the plate is analytically solved and then used for stress and deflection calculations, assuming each plate to be a simply supported beam. Results indicate that due to superposition of the induced forces between plates in a multiple-and-parallel plate array, the maximum deflection and bending stress occur at the midpoint of the outermost plate. The maximum shear stress, which is inversely proportional to plate thickness, occurs at both ends of the outermost plate.

INTRODUCTION

Direct electrical heating of a metal test section is one of the methods which have been extensively used in tests to simulate decay heating in reactor fuel elements. The required currents are quite high if a low resistance material like aluminum is used in these tests. These high currents will result in an induced magnetic force between adjacent plates that will subject the plates to bending and shear stresses. The force is attractive when currents on different plates have the same direction, otherwise, it is repulsive [1]. For a practical situation where the plates are cooled by adjacent water flow, the induced stresses can alter the geometry of the coolant channels formed between the plates and thus result in an unpredictable heat transfer pattern. Since a significant alteration to the geometry is undesirable in these tests, it is important to accurately calculate the magnitude of this effect.

CALCULATION AND ANALYSIS

Induced Magnetic Force

Based on Ampere's rule, magnetic field of a linear current 'i' is $B = k_1/d^2$, and magnetic force is $F = 2(i/d)k_2$, where d is the distance,

and k_1 and k_2 are constants depending on the units chosen [1]. In the case of a thin plate with breadth 'a' carrying current 'i' into the paper, the magnetic induction force per unit length has two components, $F_x = 2(i/a)(\alpha_2 - \alpha_1)k_2$ and $F_y = 2(i/a)\ln(\rho_2/\rho_1)k_2$, where α and ρ are the angles and distances from the point in space to the edges of the plate. Now consider two parallel wires of equal length L (cm), at a distance b (cm) apart, carrying d.c. or single phase a.c. i_1 and i_2 (ab-amperes) respectively [2], the total magnetic force (dyne/cm) on wire 2 per unit length is (see Figure 1.1)

$$\begin{aligned} F &= (\mu/\mu_0) (2 \times 10^{-7} i_1 i_2) [(1 + (b/L)^2)^{1/2} / b - 1/L] \\ &= (\mu/\mu_0) (2 \times 10^{-7} i_1 i_2) (1/b - 1/L) \\ &= (\mu/\mu_0) (2 \times 10^{-7} i_1 i_2 / b), \quad (\text{as } L \rightarrow \infty \text{ and } t \rightarrow 0) \end{aligned} \quad (1)$$

where μ represents magnetic permeability (cm-g/°C²) of medium (m) or in vacuum (v).

Taking integration twice on equation (1) with respect to breadth 'a', the force exerted between two parallel and symmetrically placed thin plates with uniformly distributed currents i_1/a and i_2/a is (see Figure 1.2)

$$\begin{aligned} F &= (\mu/\mu_0) (2 \times 10^{-7} i_1 i_2 / a^2) \int_0^a \int_0^a (b \, dz_1 \, dz_2) / (b^2 + (z_2 - z_1)^2) \\ &= (\mu/\mu_0) (2 \times 10^{-7} i_1 i_2 / a^2) \int_0^a [\tan^{-1}(z_2/b) + \tan^{-1}((a - z_2))] \end{aligned}$$

$$\begin{aligned} & \int_0^b \int_0^L f(z_2) dz_2 \\ & = \int_0^b \int_0^L f(z_2) dz_2 \\ & = (\mu_0/\mu_v) (4 \times 10^{-7} i_1 i_2 / a) \{ \tan^{-1}(a/b) (b/2a) \ln(1+(a/b)^2) \} \end{aligned} \quad (2)$$

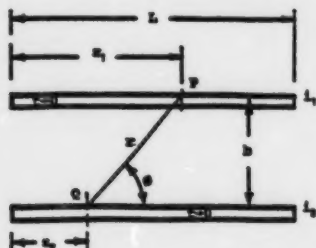
where f is the unit force between these two plates.

Plate Deflection

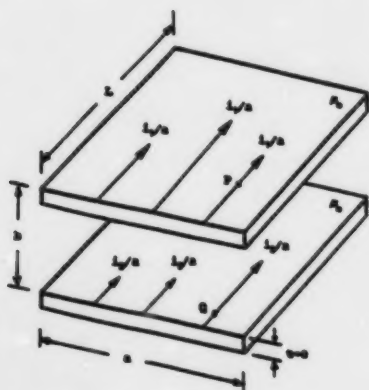
If fuel plates are assembled by swaging, as shown in Figure 2.1, the simple beam and the fixed-end beam model bound the actual end conditions on the plate and permit calculation of plate stress and deflection. For conservatism, the simple beam model which yields a larger plate deflection and normal stress is chosen in this analysis (see Figure 2.2). The deflection equation is

$$\delta = \iint (M/EI) (dx)^2 \quad (3)$$

where δ is the deflection (m), M the bending moment (N-m), E the modulus of elasticity and $E=10^7$ psi $=6.9 \times 10^4$ N/m² for aluminum, I the moment of inertia and $I=t^3/12$ m⁴ per unit length



(1) Calculation of Force between Two Equal Parallel Wires



(2) Calculation of Force between Two Equal Parallel Plates with uniformly distributed Currents

Figure 1. 2-D and 3-D Coordinates used for Magnetic Induction Force Calculations

along the direction of current, t the beam thickness (m), and x the distance (m) from beam end.

Assuming currents i_1 and i_2 (amperes) are uniformly distributed on two adjacent plates as shown in Figure 1.2, the bending moment at any point on plate 1 or 2 is

$$\begin{aligned} M &= (F/2)x - \int (fx) dx \\ &= (\mu_0/\mu_v) (2 \times 10^{-7} i_1 i_2 / a^2) \{ (ax - (a^2 - b^2)/2) \tan^{-1}(a/b) + \\ & \quad (bx) \ln(b) + (a-x) (b/2) \ln(a^2 + b^2) - ((x^2 + b^2)/2) \tan^{-1}((a-x)/b) - \\ & \quad (ab/2) \ln(x^2 - 2ax + a^2 + b^2) \} \end{aligned} \quad (4)$$

Therefore, the exact solution for plate deflection based on the boundary conditions of $\delta=0$ at $x=0$ and at $x=a$ can be expressed as

$$\begin{aligned} \delta &= [(4 \times 10^{-7}) (\mu_0/\mu_v) \{ i_1 i_2 / a \} (1/EI)] \{ (1/12a) [(3b^4 - 6b^2x^2 - x^4)/4 \tan^{-1}(x/b) + (3(a^4 + b^4) - 18a^2b^2 + 8a(3b^2 - a^2)x + 6(a^2 - b^2)x^2 - x^4)/4 \} \tan^{-1}((a-x)/b) + \\ & \quad (3ab(b^2 - a^2) + 2b(3a^2 - b^2)x - 3abx^2)/2 \ln(x^2 - 2ax + a^2 + b^2) + (b^3x) \ln(x^2 + b^2)] + \\ & \quad (1/12) [\tan^{-1}(a/b) - (b/2a) \ln(1 + a^2/b^2)] x^3 + (1/8a) [(b^2 - a^2) \tan^{-1}(a/b) + (ab) \ln(a^2 + b^2) + 13ab/6] x^2 + \\ & \quad (1/48) [3(a^2 - b^4/a^2) \tan^{-1}(a/b) - 4(b^3/a + ab) \ln(a^2 + b^2) - 4(b^3/a + ab) \ln(b)] 13ab] x + (1/16a) [(6a^2b^2 - a^4 - b^4) \tan^{-1}(a/b) + 2ab(a^2 - b^2) \ln(a^2 + b^2)] \} \end{aligned} \quad (5)$$

For reference, the maximum deflection in terms of the induced magnetic force between plates 1-2, 1-3, 1-4, 1-5, 1-7, and 1-10 at $\mu_v \approx \mu_0$, $i_1 = i_2 = 4000$ amperes, $a/t = 6.096 \text{ cm}/0.127 \text{ cm} (=2.4 \text{ N}/0.05 \text{ N})$, and $EI = 11.77 \text{ N-m}^2$ are evaluated and listed in Table 1.

TABLE 1

Maximum Deflection and Stresses of the Outermost Plates

$$\begin{aligned} \text{Plate Deflection} &= \iint (M/EI) (dx)^2 \\ \text{Plate Bending Stress} &= Mx/2I = M/S \\ \text{Plate Shear Stress} &= 3V/2A = 7/2 \end{aligned}$$

Inter-element Spacing (cm)	Deflection (cm)	Maximum Bending Stress (N/m ²)	Shear Stress (N/m ²)
b=0.254 (force 1-2)	3.51*10 ⁻³	4.89*10 ⁴	8.64*10 ⁴
b=0.762 (force 1-3)	2.77*10 ⁻³	4.60*10 ⁴	7.33*10 ⁴
b=1.143 (force 1-4)	2.36*10 ⁻³	4.28*10 ⁴	6.62*10 ⁴
b=1.524 (force 1-5)	2.01*10 ⁻³	3.91*10 ⁴	6.02*10 ⁴
b=2.286 (force 1-7)	1.43*10 ⁻³	3.26*10 ⁴	5.08*10 ⁴
b=3.429 (force 1-10)	8.86*10 ⁻⁴	2.45*10 ⁴	4.08*10 ⁴
Location	x=a/2	x=a/2	x=0 and x=a

Notes: $i_1 = i_2 = 4000$ amp, $a = 6.096$ cm, $t = 0.127$ cm, $\mu_v = 6.9 \times 10^4$ N/m², $E = 1.17 \times 10^7$ N/m², $I = 1.71 \times 10^{-8}$ m⁴.

Bending and Shear Stresses

For a plate constructed of an elastic material with a linear stress-strain relationship, the bending (normal) stress for a plate with a rectangular cross section is

$$\sigma_x = M/S \quad (6)$$

Here, σ_x and S represent, respectively, the plate bending stress (N/m^2) and section modulus (m^3) per unit length (m). For a rectangular plate, $S = 1 \cdot t^2/6$.

The maximum bending stress according to equation (6) occurs at the midpoint of the plate (i.e. at $x=a/2$), where the maximum bending moment M can be calculated using equation (4).

As opposed to bending stress, the maximum shear stress appears at both ends of the beam (at $x=0$ & $x=a$), corresponding to the maximum local shear force V , where $V = F/2 - \int (f)dx = F/2$. The general shear stress equation for a rectangular cross-sectional plate is $\tau = (V/IL) \int_0^z z dA$, at the neutral axis of z (i.e. $z=t/2$), from which the maximum shear stress can be derived as

$$\tau_{max} = 3V/2A \quad (7)$$

where τ_{max} is the maximum shear stress (N/m^2), V the maximum local shear force (N), and $A = 1 \cdot t$ the cross-sectional area (m^2) per unit length.

It is worth emphasizing that although each of the terms F , V , δ , σ , and τ is additive, that is, $F = F_{1,2} + F_{1,3} + \dots + F_{1,n}$, etc., there is no linear charge relative to distance b between plates, simply because of the appearance of transcendental functions of $\tan^{-1}(x)$ and $\ln(x)$ after the integration of the induction force ' f ' or bending moment ' fx '. For comparison, both maximum shear and bending stresses due to an induction force of 4000-amp current on each plate are calculated for values of $a = 6.096 \cdot 10^{-2} m$ ($= 2.4''$), $t = 1.27 \cdot 10^{-3} m$ ($= 0.05''$), $A = 1.27 \cdot 10^{-3} m^2$, $S = 2.69 \cdot 10^{-7} m^3$, and $I = 1.71 \cdot 10^{-10} m^4$, and the results are listed in Table 1.

CONCLUSION AND DISCUSSION

Analytical solutions of plate deflection and stresses due to the presence of induced magnetic force are obtained when electrical current is used to generate heat in a plate. Based on field theory which states that strength of the magnetic force among plates is additive, the deflection and stress on one plate can thus be evaluated in terms of forces between the specified plate and all the other plates. In a multiple-plate arrangement as shown in Figure 2.3, the inner plates always feel a smaller total force than do outer plates, due to the opposing forces exerted by plates on opposite side of the middle plate in question; therefore, the maximum deflection occurs at the two outermost plates. The reduction of inter-element spacing between the first and second plates can be described by

(Space Reduction)_{1,2}
 $= \delta$ of plate 1 - δ of plate 2
 $= \delta$ due to forces between plates 1-2 and 1-7
 $\approx \delta$ due to force between plates 1 and 2 only
 $= \delta$ due to force between any 2 adjacent plates

Based on the same analysis, the reduction of the second inter-element spacing can be computed by

(Space Reduction)_{2,3}
 $= \delta$ of plate 2 - δ of plate 3
 $= \delta$ due to forces between plates 2-4 and 2-7
 $\approx \delta$ due to force between plates 2 and 4 only
 $= \delta$ due to force between plates $n-1$ and $n+1$

Because of symmetry, the middle plate shows no deflection, as long as a constant and uniform current heats all plates.

From equation (6), the maximum bending stress relative to an induced force resulting from a 4000-amp current on each of the 2nd through 5th plates is $1.77 \cdot 10^7 N/m^2$, occurring in the middle of the 1st plate. Based on equa-

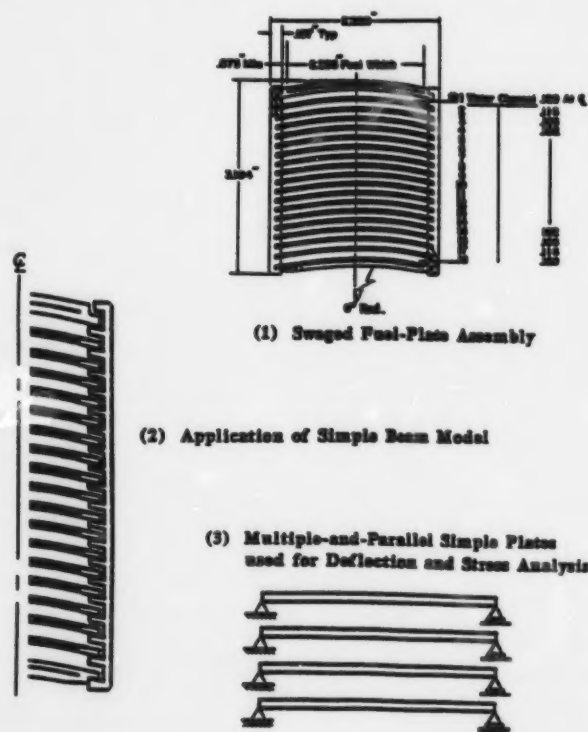


Figure 2. Simple Beam Model used for Plate Deflection and Stress Calculations

tion (7), the maximum shear stress on the same plate arrangement experiencing a 4000-amp current would be $2.87 \cdot 10^{-5} \text{ N/m}^2$, at either end of the 1st plate. In contrast to the stresses on all inner plates where the induced force can be partially or even totally (middle plate only) cancelled due to 'symmetry', maximum bending and shear stresses are evaluated respectively from the total bending moment and total shear force between plate 1 and all other plates, since induced magnetic force on the outermost plates is the summation of all individual forces when current flow on all plates is unidirectional.

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TECHNICAL SESSION 2

MATERIALS-FABRICATION

Chair: S.D. Bhole

REDUCING AND SOLVING LATERAL VIBRATION PROBLEMS IN PUMP MOTOR SETS.

David Grier, P.Eng., Director
Tony Kaminski, P.Eng., Ph.D., Senior Design Engineer

Equipment and Machinery Development Division
Saskatchewan Research Council
15 Innovation Blvd., Saskatoon, Saskatchewan S7N 2X8
Telephone: (306) 933-8131; Fax: (306) 933-7896

ABSTRACT: This paper provides design and maintenance personnel with information to aid in reducing, detecting and dealing with vibration problems in high speed pump motor sets. A basic understanding of vibration is provided. Methods to reduce vibration problems through proper design, specification, procurement, installation and commissioning procedures are outlined. The use of vibration monitoring and analysis instrumentation and software is described. The keys to implementing a successful vibration monitoring program are listed.

1.0 INTRODUCTION

The Canadian Electrical Association (CEA) and SaskPower commissioned a study to be conducted by SRC to document the state of the art of the methods to reduce, detect and correct vibration problems in high speed pump motor sets. This paper is a brief summary of this work. Copies of the complete report are available through CEA.

2.0 VIBRATION BASICS

Definition: Vibration is simply the motion or oscillation of a body about a reference position. It is the response of a system to an excitation force that has been applied either internally or externally and is changing in either magnitude or direction.

Vibration Causes: Vibration in pump motor sets is caused by mechanically, electrically, and/or hydraulically induced forces.

Vibration Parameters: A machine's condition can be determined by measuring its vibration parameters. The more important of these parameter are:

- Frequency
- Displacement
- Velocity
- Acceleration
- Phase

Vibration frequency is the measure of the number of complete cycles that occur in a specified period of time. It is usually expressed in cycles per minute (CPM) or in hertz (Hz) or cycles per second.

The total distance travelled by a vibrating part is referred to as the **peak-to-peak displacement**. It is usually expressed in mils (1 mil is 1/1000 of an inch) or micrometers. The vibration amplitude is one half the peak-to-peak displacement.

The **velocity**, or rate of change of displacement, is constantly changing throughout each cycle. Therefore, the highest, or "peak", velocity is measured. It is expressed in inches per second peak or millimetres per second peak. The velocity of the part is zero at the extreme limits of the part's travel when the displacement is at its maximum value.

Each time the part comes to a stop at the limit of travel it must speed up, or accelerate, again. **Acceleration** is the rate of change of velocity. Acceleration is generally maximum at zero velocity and maximum displacement. It decreases as the velocity increases. When the vibration moves through the mid point of its travel, the velocity is maximum and the acceleration is zero. On the other side of the midpoint the acceleration becomes negative as it begins to decelerate the part as it moves toward the other limit of travel. Acceleration is normally expressed in "g's" peak, where 1 g is the acceleration produced by the force of gravity.

Phase is the position of a vibrating part at an instance with reference to a fixed point or another vibrating part. Phase measurements offer a way to compare one vibration motion with another or to determine how one part is vibrating relative to another part. They are essential for detecting resonance, looseness, misalignment, some types of unbalance and for determining mode shapes.

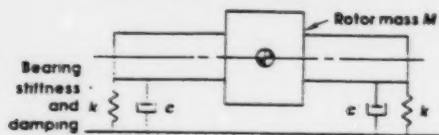
Rotor Dynamics: All rotating machinery will produce vibration. Forced vibration is the vibration caused by an external force such as a shaft imbalance or torque variations. The frequency of the driving force will dictate the frequency of this vibration. Free vibration is the vibration that occurs when a body vibrates "naturally" in the absence of a continued external force. The frequency or rate at which a body vibrates freely is the natural frequency.

Natural frequencies occur at many frequencies, with the lowest natural frequency having the largest displacements. Resonant frequency is the frequency at which the natural and driving frequencies coincide. In rotor dynamics, the term **critical speed** is used to describe the resonant frequency that occurs when the rotational speed (driving force frequency) coincides with the natural frequency. As there is more than one natural frequency, there is more than one critical speed.

At critical speeds, the shaft becomes dynamically unstable, resulting in excessive deflection or vibration. Shafts will take on different shapes, called **mode shapes**, at each critical speed as shown in Figure 1. The

critical speed is affected by the rotor stiffness, the rotor mass and the stiffness of the rotor support structure (i.e. the bearings). It is important that machinery not be operated within 20% of a critical speed. A designer must consider that shaft critical speeds are not fixed values because bearing stiffness reduces over its life. Avoiding critical speeds becomes even more of a challenge when variable speed drives which permit the machine to be operated at speeds including critical speeds are introduced.

A rotor-bearing system can...



...vibrate at more than one critical speed

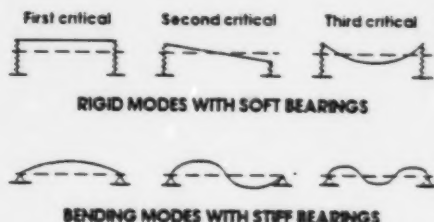


Figure 1: Critical Speed Mode Shapes

Unbalance: Unbalance is the unequal distribution of weight of a rotating body about the rotating axis. In other words the centre line of the mass or principal axis of the rotor shaft is not the same as the centre line of rotation. The magnitude of the force created by unbalance depends on the amount of unbalance and the rotor speed.

As a general rule, rotors operating below 70% of the first critical (lowest critical speed) are considered to be rigid. Balancing at any speed within the operating range results in a balanced rotor at any other speed. Shafts operating above 70% of the first critical are considered flexible because rotor bending due to the forces of unbalance is significant. A flexible rotor balanced at one speed may not be balanced at another speed. Flexible rotors may require multi-plane balancing rather than single or two plane balancing.

There are a number of balancing standards available. The International Standards Organization (ISO) has developed balancing standards for various grades of equipment. A more stringent standard is the American Petroleum Institute's standard for centrifugal pumps (API 610) would be the better choice for critical equipment like pump motor sets.

3.0 VIBRATION PROBLEMS IN PUMP-MOTOR SETS

Electric Motor Vibration: Vibrations in an electric motor can be caused by mechanical or electrical forcing functions. Mechanical forcing functions include:

- unbalance
- resonance (operation near a critical speed)
- misalignment

- bent shaft
- looseness of windings, base, bearings
- defective bearings
- rubs
- thermal bow of the rotor.

Electrical forcing functions can be:

- electrical discharge within stator bar insulation
- slot discharge between stator bar insulation and stator core
- end winding surface discharge
- broken conductor discharge
- unbalanced phase resistance
- loose stator laminations
- electrical misalignment

A number of tests and calculation are required to diagnose motor problems. It is important to determine whether a motor vibration problem is electrical or mechanical. It is often possible to detect electrical problems by taking electrical motor data during start-up or shut-down.

Table 1 shows a summary of electric motor problems identified in a CEA study conducted by Ontario Hydro (Ref. #12). The motor sub-assembly consisting of the bearings, seals and lubrication system were the major causes of motor failures. A much larger study conducted by IEEE had similar results. The data showed that motor bearings were the major failure area. The failures were attributed to vibration and poor lubrication caused by inadequate maintenance.

TABLE 1
ELECTRIC MOTOR SUB-ASSEMBLY FAILURES

Failures by Motor Sub-Assembly	
Sub-Assembly	Percent
Frame and Bracket Assembly	3
Bearings, Seals & Lube System	45
Stator Assembly	12
Rotor Assembly	10
Accessories & Instrumentation	27
Unknown	3

Centrifugal Pump Vibration: Vibrations in a centrifugal pump can be caused by mechanical or hydraulic forcing functions. Mechanical forcing functions include:

- unbalance of rotor including the coupling
- rotor instability due to excessive wear
- ring clearance
- bent shaft
- improper assembly
- defective parts
- excessive deflections due to thermal growth and weight of piping
- defective bearings or seals

Hydraulic forcing functions include:

- cavitation
- recirculation
- hydraulic unbalance or flow disturbances
- impeller to volute/diffuser interaction
- leaking valves

All pumps vibrate to some extent. Vibration can be caused by some of the more obvious mechanical problems such as a bent shaft, bad bearings or loose coupling. A less obvious mechanical problem is when the weight of the piping bends the pump housing relative to the shaft resulting in a reversing force on the shaft or even a rub. "Softer" bearings, that

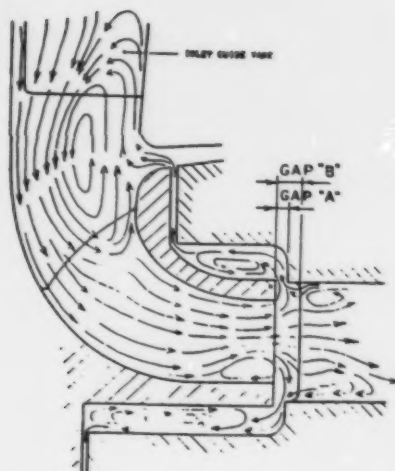


Figure 2:
Secondary Flow Pattern at Off-Design Flow

allow too much shaft deflection, will result in more vibration. "Stiffer" bearings will reduce vibrations and change the critical speeds of the rotor.

Vibrations caused by hydraulic forces vary significantly, depending upon where the pump is operating relative to its design flow. As the flow volume is decreased through an impeller, a point where the flow between the impeller blades reverses itself is reached. At this point, called the surge point, large pressure pulsations are experienced. Flow recirculation can also happen at low flows, also causing pulsations (see Figure 2). Both these types of flow separation conditions will cause vibrations.

The design of the impeller and diffuser or volute and their relationship has a significant impact on vibration. Pressure pulses are created by each impeller blade. Higher pressure pulses will be created by impellers with few blades. Multi-stage in-line pumps, such as those used in feed water pumps, should not use impellers with the same number of blades or multiples of that number to avoid simultaneous pressure pulses, which will result in higher amplitude vibrations. Four blade impellers should not be used. In pumps employing diffusers, impeller/diffuser vane combinations such as 6-8, 6-9, and 6-10 need to be avoided. If the impeller back plate is too thin, it will deflect resulting in pulsing flows and vibration.

Turbulent flow will cause vibration. Turbulence can be created within the pump by not centering the impeller/diffuser hydraulic channel or by not providing adequate increase in diffuser width relative to impeller width to permit all the water leaving the impeller to enter the diffuser. Reducing the gap between the impeller and the diffuser or volute will increase the efficiency of the pump as well as the magnitude of the pressure pulses (see Figure 3).

Pump cavitation is another problem that will appear as a vibration. The high velocities experienced at high flow rates at the impeller inlet vane edge cause localized areas of low

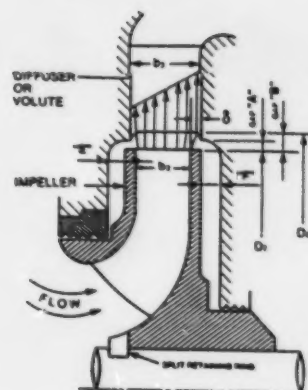


Figure 3:
Impeller to Diffuser/Volute Size Relationship

pressure. Water vapour bubbles are formed as the pressure falls below the saturation pressure of the water. These bubbles implode or collapse rapidly when the local pressure again rises above the saturation pressure. This will cause the removal of material from the pump impeller. As the material erosion is not uniform, the impeller becomes unbalanced, resulting in vibration.

API 610 states that the pump is to operate within + or - 10% of its rated speed and capacity. Hydraulic Institute Standards (HIS) state that pumps must be operating in a non-cavitating and non-separating condition. This usually means between 60% and 110% of the best efficiency flow.

4.0 PRACTICES FOLLOWED TO REDUCE VIBRATION

A good understanding of vibration will result in a great deal of preventative maintenance. Although vibration cannot be completely eliminated, it can be significantly reduced and controlled with the implementation of a number of practices. There are quite a number of industry specifications that will assist in minimizing vibration problems. Those referred to in the preparation of this paper are listed in the references.

Induction Motor: There are a number of motor design features that can be specified that will have an impact on vibration reduction. A review committee of the IEEE has recommended that the following be incorporated into the ANSI/IEEE Standard (Ref. #7):

- TEFC enclosures
- Maximum 80°C rise at 1.0 service factor
- Contamination protection for bearings and grease reservoirs
- Three year continuous L-10 bearing life
- Maximum bearing temperature rise of 45°C (two pole motor, 50°C)
- Cast iron frame construction
- Non sparking fan
- Single connection point per phase in terminal box
- Maximum sound level of 90 DBA
- Corrosion resistant paint on internal joints, surface and hardware.

Ontario Hydro recommends the following to increase motor reliability based on their analysis of motor failures:

- Use Class 150 insulation for applications requiring Class 130
- Allow at least 10% margin between design load and motor rating
- Insist on full load test to get full load winding temperature
- Monitor temperatures in the hottest areas of the stator windings.

Electrical motor manufacturers typically have strong electrical engineering groups but are much weaker in the mechanical engineering area. Critical speed calculations should be conducted (or required of the manufacturer) on large 2-pole horizontal and 4-pole vertical motors because these motors tend to have higher vibration levels. Consideration must be given to the fatigue caused by thermal or mechanical stresses primarily in 2-pole motors. T-shaped rotor bars should not be used in high inertia applications. The natural frequency of the cooling fan must be above 140 Hz.

Attention must be paid to bearing clearances, rotor critical speeds, and lubrication viscosities in hot and cold conditions to reduce vibration due to unbalance. Bearing problems can be caused by deterioration of insulation due to vibration, especially in 3600 RPM motors; oil leakage; dust and/oil contamination and inadequate clamping of insulated steel bearings. Steel sleeve bearings should be specified for 3600 RPM motors rather than cast iron sleeves.

Pump: A number of design features that can be specified that will minimize pump vibration problems have been listed below.

- Specify labyrinth seals rather than floating ring seals for better bearing protection
- Specify tilting pad journal bearings (or at least wide land journal bearings) to increase rotor stiffness
- Specify an impeller with an odd number of vanes (especially stay away from 4 vane impellers)
- Specify proper impeller-to-diffuser/volute vane combinations
- Specify adequate diffuser width to impeller width ratio to minimize problems of axial alignment
- Obtain flow versus instability data and select a pump that will deliver the flow requirements in the most stable portion of its operating range
- Specify low journal bearing clearances
- Specify impeller to diffuser or volute radial clearances of 3 to 5% of diameter for diffuser pumps and 6 to 10% for volute pumps
- Specify adequate impeller back plate thickness (usually 3/8" to 1/2")
- Specify optimized impeller-to-diffuser/volute gap geometry
- Specify a shaft surface suitable for use with a proximity probe
- Specify the installation of temperature and vibration transducers at the factory
- Specify net positive suction head available to be 1.5 to 1.8 times the net positive suction head required to prevent cavitation

Equipment Procurement: The key to procurement is good system design and the development of specifications that will result in the purchase of components that perform as the designer expected. Both reliability and performance of pumps and motors have to be evaluated. For example, high efficiency pumps have reduced clearances and are much more prone to rubs and seizures. Therefore some specifications state that no credit will be given for efficiencies above 85% at the best efficiency point.

Procurement specifications should outline tests to be conducted at the manufacturer's shop for the pump and the motor. The buyer should witness the test under varying load conditions. Motor temperatures, vibration levels and efficiencies need to be measured. The Hydraulic Institute's Centrifugal Pump Standard 1.6, 1988 states that vibration, gland seal water flow, balance line water flow, pressure and oil temperature measurements should be taken at 8 equally spaced flows between the best efficiency point and the minimum flow point. Warm-up and start-up procedures must be clearly outlined and type and value of parameters to be measured specified. Typical parameters include allowable temperature differential between casing and fluid, allowable temperature differential between top and bottom of pump casing, and allowable rate of temperature rise. API 610 outlines inspection, testing, vendor data and drawing requirements.

Equipment Installation and Commissioning: Even with the best design, manufacturing and procurement procedures, vibration can still result if the machinery is not installed and commissioned properly. The following information must be acquired from the vendor:

- foundation requirements
- grouting and levelling procedure
- maximum loads at pump flanges
- cleaning and flushing procedure for piping
- flushing procedure for oil lube piping
- shaft alignment requirements
- commissioning tests required

The following commissioning procedures should be followed:

- Operate the motor without being coupled to the pump until the bearing temperatures become normal.
- Then take baseline vibration data on the motor.
- Install the coupling and operate the pump at design flow until the temperatures have stabilized.
- Acquire vibration data for pump and motor. If the vibration levels are unacceptable, identify the problem. Common problems include misalignment and excessive forces on the pump flanges.
- After acceptable vibration levels are obtained at design flow, gather baseline data at design flow and other off design flow conditions.

Baseline data should include motor currents, motor and pump vibration, and motor and pump temperatures for various pump operating points. This information is vital to diagnose future problems.

Maintenance Procedures: Mechanical components, such as bearings and gear couplings, require regular maintenance to minimize wear and to keep vibration under control. Regular monitoring of vibration levels, motor current, motor and pump temperatures, and analysis of lubricating oil will enable maintenance personnel to predict the timing and type of repair required. This approach will maximize the time between maintenance shutdowns while greatly reducing the chances of in service failure.

5.0 VIBRATION MEASUREMENT AND ANALYSIS

Vibration Measurements: Vibration measurements include amplitude, frequency and phase angle. Amplitude can be measured in displacement, velocity and acceleration and is the primary indication of a machine's condition. Frequency and phase angle are of major importance in diagnosing the cause of a problem from vibration data.

A vibration meter, or analyzer, is most often used for collecting vibration data. This unit will receive and store information sent to it from amplitude, phase angle, time, rotation speed and other transducers. A number of types of measurement are required to provide the information necessary for vibration problem diagnosis.

Amplitude versus frequency spectrum measurements are the most useful type of measurement since they identify primary frequencies that contribute to the overall vibration level as shown in Figure 4. Advanced sampling analyzers are required to take these measurements. In the more common FFT analyzer, a digitized signal is processed by a computer algorithm to yield the discrete Fourier transform of the signal over a selected frequency range. The Fourier transform contains the amplitude and phase information as a function of frequency. In a time-compression analyzer, a digital memory is used to "speed up" a signal which is then fed into a high-frequency analog spectrum analyzer. This type of analyzer can sweep at a much higher rate than a low frequency analyzer of comparable band width. Sampling analyzers are suitable for analysis of both transient and steady-state vibration signals.

FREQUENCY BANDS OF INTEREST AND AMPLITUDE LIMITS

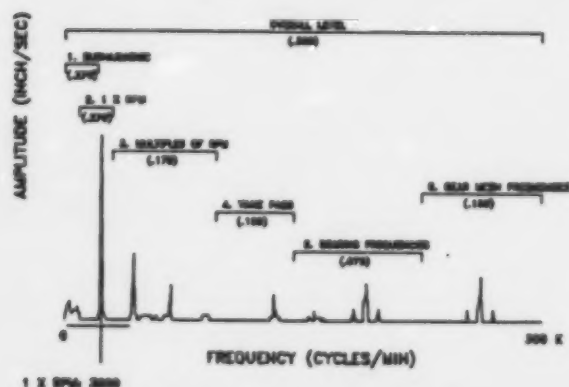


Figure 4. Amplitude Versus Frequency Spectrums (FFT Analysis)

Vibration Analysis: Vibration frequency data can be divided into three major components relative to shaft frequency or RPM. These are the:

- **Synchronous components**, which are vibration frequencies that are multiples of shaft speed ($N \times \text{RPM}$, where N is an integer). Synchronous components of vibration up to 8 times the shaft speed can indicate unbalance, misalignment, bent shaft, looseness, pitch line runout, cavitation, blade or vane pass or non-linear motion. Multiples higher than 8 can indicate higher blade or vane pass, harmonics of lower multiples, gear mesh and motor slot pass.

- **Sub-synchronous components**, frequencies less than the shaft speed. Sub-synchronous components can be an indication of oil whirl, oil whip, another machine in the area, rubs, anti-friction bearing loose in housing, cage frequency of anti-friction bearing, defective gear tooth repeat frequency, or surge (flow reversals inside an impeller).

- **Non-synchronous components**, frequencies that are greater than the shaft speed, but not injector multiples of the shaft speed. These frequencies can be an indication of anti-friction bearing problems, system or structural resonances, hydraulic or electrical forcing functions.

Vibration data can be collected continuously or with portable analyzers. Commercially available software can process the data, compare it with base line information, plot trends and summarize results. Some of the software has detailed information such as bearing frequencies for specific bearing designed to assist in the analysis of data. A number of expert systems have recently been developed to assist in data analysis. These systems are capable of screening large amounts of vibration data, directing the analyst to problem areas and providing lists of probable causes.

6.0 TECHNOLOGY TRANSFER

Creating a Successful Vibration Monitoring/Analysis Program: An employee of Dayton Power stated that the result of a \$140,000 investment for training and vibration equipment has saved this utility \$7 million over 8 years. Vibration analysis is an essential component of any successful predictive maintenance program. The investigators contacted a number of utilities in the United States that have implemented successful vibration monitoring/analysis programs and identified the following characteristics of a successful program.

Most utilities doing vibration analysis have a trained full time technician at each plant to obtain and analyze vibration data. In addition, some of these people do balancing and alignment work. These technicians are supported by a central engineering group or an outside consultant who typically do advanced analysis and assist plant personnel in various areas including trouble shooting and training. If a utility has many plants located around a large metropolitan area then one corporate group does all the vibration work for the plants on a regular basis.

Most utilities obtain vibration data on numerous pieces of equipment using portable vibration analyzers which do FFT analysis and

display frequency spectrums. Amplitudes and often phase angles are monitored. Portable analyzers made by CSI, TEC, Palomar or IRD are commonly used.

Vibration data is usually taken monthly on equipment when operated at rated load. Plant personnel will create a schedule for vibration measurement that is followed rigorously. Usually about 10 measurements are recorded per machine. A maximum of 15 sets of data are stored in a computer at anytime and older data is dumped or in some cases, such as with nuclear equipment data may be stored for a number of years. Continuous vibrating monitoring is normally done only on turbines and generators and occasionally on critical pumps such as boiler pumps.

Most of the vibration data is obtained in the form of velocity with some displacement data and in special situations acceleration levels are recorded. Hand held or magnetically mounted sensors are commonly used to obtain housing data with wooden shaft sticks or proximity probes used to obtain shaft data. Phase angle data is also obtained for some machinery.

Data collected by portable analyzers is dumped into a host computer where it is analyzed using commercially available software and compared with baseline data. Trends are plotted and results of analysis are printed out. Machinery requiring balancing, alignment or servicing is identified. Vibration data from plants is normally made available to the corporate office using modems.

In addition to normal FFT analysis various other software programs including expert systems are being considered or evaluated by utility personnel. It is estimated that over 90% of the machinery problems will be identified by using FFT analysis. Software is generally purchased from suppliers, not developed in house.

7.0 CONCLUSIONS

Significant reduction in vibration problems with high speed pump motor sets can be realized with proper design, specification, procurement, installation, commissioning, operation and maintenance procedures. Predictive maintenance programs that employ vibration monitoring and analysis will lead to the detection, diagnosis, and solving of vibration problems before they can cause expensive down time. Good procedures to deal with vibration will result in significant savings, easily justifying the cost of their implementation.

A good understanding of vibration basics and the potential savings to a company are necessary in many job functions. Designers, purchasers, construction supervisors, operators and equipment maintenance personnel all affect the impact vibration will have on a utility. Readily available, user friendly monitoring and analysis equipment and software make it very possible to transfer this technology to the working level, thereby ensuring a successful program for dealing with vibration.

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A FAILURE CRITERION FOR SUPERPLASTIC DEFORMATION IN CERAMICS

A. Prakash and S. D. Bhole
Industrial Systems Engineering
University Of Regina
Regina, Canada, S4S0A2

ABSTRACT

This paper presents the development of a failure criterion to predict failure stresses during superplastic deformation of ceramics. The formulation is based on constitutive relationships which describe the flow stress in the deforming ceramic material during superplastic deformation. It has been assumed that the constitutive relationships which describe the flow stress also govern the stress in the deforming material after the ultimate stress point. A power-law has been formulated relating stress to strain. It has been assumed that this relationship holds even after the ultimate stress point. In the present paper it has been shown how the failure criterion can be utilized to select deformation parameters for bulge-forming operations. 20%Al₂O₃-YTZ has been taken as an example to illustrate the development of the failure criterion.

NOMENCLATURE

$\dot{\epsilon}$	strain-rate, s ⁻¹
A	a constant, 3×10^8 for 20% Al ₂ O ₃ -YTZ
d	instantaneous grain size
d ₀	initial grain size
q	a constant, 1.5 for 20% Al ₂ O ₃ -YTZ
σ	equivalent stress
n	strain-rate sensitivity factor, 2 for 20% Al ₂ O ₃ -YTZ
Q	activation energy, 590 KJ/mol for Al ₂ O ₃ -YTZ
T	absolute temperature
e _f	percentage elongation to failure
P	a constant, 1.4×10^4 for 20% Al ₂ O ₃ -YTZ
f	a constant, -0.33 for 20% Al ₂ O ₃ -YTZ
ϵ_f	effective strain to failure
P _f	a constant, 1.4×10^2 for 20% Al ₂ O ₃ -YTZ
σ_f	effective stress at failure
t _f	time to failure
$\bar{\sigma}$	equivalent stress during bulge-forming
σ_θ	hoop stress
σ_ϕ	meridional stress
d $\bar{\epsilon}$	equivalent strain during bulge-forming
p	bulging-pressure
r _{max}	maximum radius of the bulged section
ρ_1	radius of curvature of the bulged section
t	wall thickness of the bulged section
r ₀	initial radius of the tubular blank
F	bulging force

INTRODUCTION

Superplasticity is usually defined as the ability of certain materials to undergo extraordinary tensile deformations (Nieh and Wadsworth, 1990). The first detailed investigation of the superplastic effect is generally attributed to Pearson in 1934 (Machara and Langdon, 1990), when an elongation of 1950% was achieved in an aged Bi-44% Sn eutectic alloy. However, intense scientific and commercial study of superplasticity started only after about 1960. In the ensuing years, it has been recognized that the superplastic process has many potential applications in the forming industry. As a result, considerable effort has been directed to the development of new superplastic materials and forming techniques.

Until recently most of the work in superplasticity has concentrated on metallic alloys. However, recent advances in the field of ceramic processing technology have greatly improved the quality of ceramic powders and ceramics are now being studied extensively for superplasticity. Superplasticity in a ceramic material was first reported in 1986 when Wakai et al. (Machara and Langdon, 1990) recorded tensile elongations of over 100% in an Yttria-Stabilized Tetragonal Zirconia (Y-TZP). Since then attempts have been made to use ceramic materials in forming operations. Some structural parts have been formed superplastically using 3Y-TZP and Al₂O₃ (Nieh et al., 1991). Superplastic ceramics are an alternative to metallic materials that offer the potential benefit of forming net shape or near net shape parts for high temperature applications.

For the development of superplastic forming (SPF) of ceramics it would be useful to formulate and study the constitutive relationships and failure criteria. A failure criterion is required to predict the extent to which these materials can be deformed during the forming operation.

This paper presents a failure criterion to predict the failure stresses during superplastic deformation of ceramics taking 20% Al_2O_3 -YTZ as an example. Further, the developed criterion is used to select the forming parameters during bulge forming.

SELECTION OF MATERIAL FOR THE STUDY

There are two types of superplasticity :

1. Transformation Superplasticity refers to the temperature cycling of a material through a phase change.
2. Structural Superplasticity refers to attaining superplastic tensile elongations without a phase change and under conditions of constant temperature.

Both types of superplastic behavior have been observed in a number of different ceramics, which can be classified into :

1. Monolithic Ceramics : eg., Yttria-Stabilized Tetragonal Polycrystal (Y-TZP), Al_2O_3 , Hydroxyapatite, β -Spodumene Glass.
2. Composite Ceramics : eg., Al_2O_3 -YTZ, 15% SiC-Si $_3\text{N}_4$, 20 vol. 20% Fe-Fe $_3\text{C}$.

The structural prerequisites for superplasticity are reasonably well understood (Machara and Langdon, 1990), viz, a small grain size, a high testing temperature and a low strain-rate. The first two requirements are generally incompatible with one another because at high temperatures rapid grain growth occurs and leads to a degradation in the superplastic properties of ceramic materials. However, superplastic ceramics at high temperatures still offer the advantage of finer grain size over superplastic metals.

As opposed to other ceramics, Al_2O_3 -YTZ shows a fairly constant strain-rate sensitivity value in the temperature range from 1250°C to 1650°C, which has been identified as the optimum temperature range for superplastic forming processes.

20% Al_2O_3 -YTZ composite contains (by weight) 4.0% Y_2O_3 , 20.0% Al_2O_3 , 0.014% SiO_2 , 0.005% Fe_2O_3 , 0.004% Na_2O , balance ZrO_2 . Its microstructure after thermal etching at 1450°C consists of equiaxed ZrO_2 and Al_2O_3 grains, both of which are about 0.5 μm . Grain size has a significant effect on superplasticity. As suggested by Nieh and Wadsworth (1990), the elongation to failure of the material decreases, and flow strength increases with increasing grain size.

Although extensive studies have been conducted to study the effect of grain size and temperature on the flow stresses during superplastic deformation, the effect of cavitation and intergranular fracture have not been studied on the superplastic behavior of 20% Al_2O_3 -YTZ. Deformation at elevated temperature leads to extensive nucleation and the growth of voids in the grain boundaries, and the material fractures prematurely due to intergranular fracture. Deformation in 20% Al_2O_3 -YTZ is primarily through a grain-boundary sliding mechanism. Sliding at a grain boundary which contains inclusions can concentrate the tensile stress at the interface at which the inclusion and matrix meet. This leads to nucleation of voids in 20% Al_2O_3 -YTZ causing premature fracture.

CONSTITUTIVE RELATIONSHIPS

As proposed by Nieh et al. 1991, superplastic flow in ceramics is a diffusion controlled process and the strain-rate $\dot{\epsilon}$ can be expressed as :

$$\dot{\epsilon} = A d^{-n} \sigma^n \exp(-Q/RT) \quad (1)$$

The values of the constants and activation energy vary according to the microstructure, specific flow or diffusion law, and, sometimes, impurity content in the material. Superplastic ceramics have a fine microstructure which is thermally unstable and undergoes grain growth during superplastic deformation. Both static and dynamic grain growths have been observed during superplastic deformation of ceramics.

In Y-TZP, which is one of the most widely studied superplastic ceramics, the static and dynamic grain growths can be expressed in the form :

$$d^l - d_0^l = kt \quad (2)$$

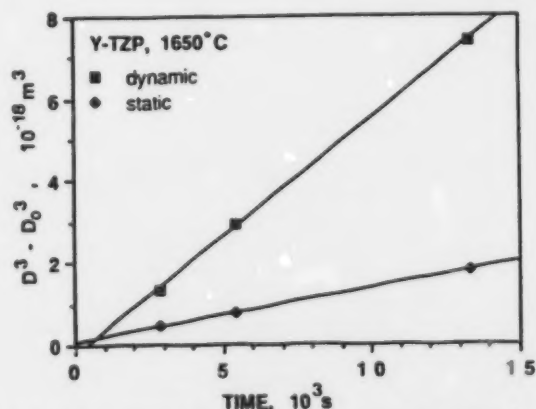
Experimental investigations have revealed that the value of l is the same (equal to 3) for both static and dynamic grain growths. The value of k depends primarily on the temperature and grain boundary energy. The value of k for dynamic grain growth during the deformation of Y-TZP at 1650°C has been found to be 0.8391 (Nieh and Wadsworth, 1990) as shown in Figure 1. Since it has been observed that the grain growth of ZrO_2 in Al_2O_3 -YTZ is similar to that in Y-TZP, the same relationship can be used to describe the grain growth of ZrO_2 in 20% Al_2O_3 -YTZ.

It has been observed (Langdon and Wadsworth, 1990) that during the superplastic deformation of 20% Al_2O_3 -YTZ, the ZrO_2 grains grow faster than Al_2O_3 grains. So ZrO_2 grains reach the critical grain size earlier than Al_2O_3 grains. Hence, it would be reasonable to focus on the grain growth of ZrO_2 .

The following equation gives the flow stress during the superplastic deformation of 20% Al_2O_3 -YTZ :

$$\dot{\epsilon} = A d^{-1.5} \sigma^2 \exp(-Q/RT) \quad (3)$$

The constant, A , has been found to be 3×10^9 for Al_2O_3 -YTZ. This constitutive relationship satisfactorily describes the effect of grain size and temperature on the stresses in the deforming material.



Slope Of Dynamic Growth Curve, k is 0.8391

Figure 1: Plot Of Grain Growth vs Time in Y-TZP

The formulation of the failure criterion is based on the assumption that stresses follow equation 3 even after the ultimate stress point. Equation 2 can be incorporated in equation 3 to describe the effect of grain growth on the equivalent stress as follows :

$$\dot{\epsilon} = A \times [kt + d_0]^{-0.5} \sigma^2 \exp(-Q/RT) \quad (4)$$

This equation can be modified to give the following expression

$$kt = \left[\frac{A\sigma^2 \exp(-Q/RT)}{\dot{\epsilon}} \right] - d_0^2 \quad (5)$$

which indicates that superplastic deformation is a time-dependent deformation process. With the passage of time, at a constant strain-rate and temperature, the stresses in the deforming material increase. Figure 2 (Wakai, 1987) shows a plot of effective-stress versus effective-strain for Al_2O_3 -YTZ at different strain-rates.

Based on the data reported in the literature, a Power-Law of the form

$$\sigma = B\epsilon^m \quad (6)$$

was formulated to express the relationship between stress and strain. As expected, the value of m was found to be almost constant (0.035) at different strain-rates. The value of B was found to depend on strain-rate. A plot of B versus strain-rate is shown in Figure 3. It has been assumed that the Power-law holds even after the Ultimate Stress Point. It has been observed that stresses in the deforming material fall after the ultimate point. However, the assumption that the relationship which describes the flow stress also governs the stresses after the ultimate stress point gives stresses which are higher than the actual stresses developed in the material. The proposed failure criterion, in this sense, is an upper-bound approach.

A strain rate of $1.02 \times 10^{-4} s^{-1}$ seems to be the most favorable strain rate giving largest elongations. For this value of strain-rate, $B = 15$ MPa.

FAILURE CRITERION

The Yield-Criterion for any material predicts the stresses at the onset of yielding in the material. During forming operations it is also important to know the limit to which the material can be deformed. Ordinary materials are normally deformed to the Ultimate Stress point. Any deformation beyond the Ultimate Stress point lies in the instability zone. Superplastic ceramics, however, can be deformed considerably even beyond the Ultimate Stress point. A failure criterion is thus desired to get an estimate of the failure stresses in the deforming ceramic material. A formulation of this criterion is presented in this section.

The elongation to failure for tensile tests can be described by a phenomenological relationship as suggested by Woo-Jin Kim et al. (1991).

$$\epsilon_f = P \left[\frac{\dot{\epsilon} \exp(Q/RT)}{A} \right]^f \quad (7)$$

This can be expressed in the form of effective strain as

$$\epsilon_f = \exp \left[\frac{P \dot{\epsilon} \exp(Q/RT)}{A} \right]^f \quad (8)$$

The Power-Law mentioned earlier (Equation 6) can be incorporated into this to get the critical stress as follows :

$$\sigma_f = B \left[\exp \left(\frac{P \dot{\epsilon} \exp(Q/RT)}{A} \right) \right]^m \quad (9)$$

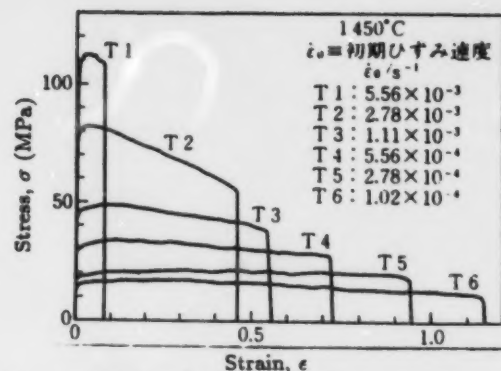


Figure 2: Plot of Effective-Stress Versus Effective-Strain

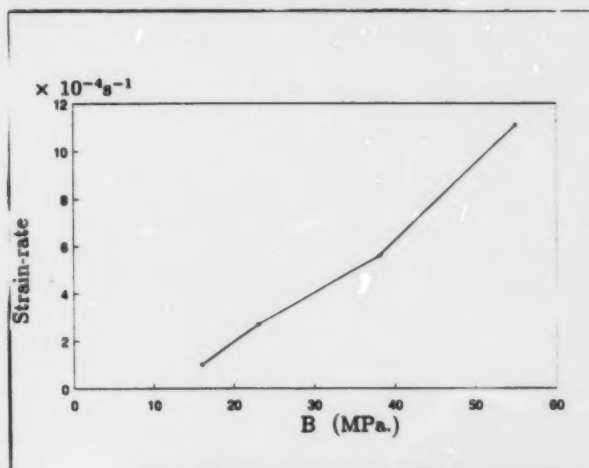


Figure 3: Plot Of Strain-Rate vs B

It would be desirable to keep the stresses below this level during superplastic forming. This expression can also be used to find the critical grain size at which failure occurs. Using the constitutive relationship (Equation 3) the critical grain size can be expressed as :

$$d = \left[\frac{AB^2 \left[\exp \left(\frac{P \dot{\epsilon} \exp(Q/RT)}{A} \right) \right]^f \exp(-Q/RT)}{\dot{\epsilon}} \right]^{\frac{1}{2m}} \quad (10)$$

By using the grain growth law, the time to fracture can be expressed as :

$$t_f = \frac{1}{k} \left\{ \left[\frac{AB^2 \left[\exp \left(\frac{P \dot{\epsilon} \exp(Q/RT)}{A} \right) \right]^f \exp(-Q/RT)}{\dot{\epsilon}} \right]^2 - d_0^2 \right\} \quad (11)$$

SUPERPLASTIC FORMING

Utilization of the superplastic properties of ceramic materials is now viewed as a technological breakthrough since it offers an opportunity to design and fabricate monolithic structures. Weight savings of approximately 50% have been achieved on parts redesigned to take advantage of the superplastic forming (SPF) process. Superplastic forming offers designers the freedom

to design complex, monolithic parts and provides increased efficiency through weight reduction resulting from reduced number of details and elimination or reduction of fasteners.

Bulge Forming

Tube bulging is a forming operation that can be employed to produce many useful components such as corrugated cylindrical tubes and bellows, tees and branched tubes, differential cases, stepped diameter tubular shafts, bicycle hubs, nozzles, hose couplings and missile venturi sections. The starting point for the production of these items is a tubular blank. Bulging is an efficient method of producing these items since it involves a lesser number of operations than most forming operations. Figure 4 schematically shows the bulge-forming operation (Thiruvandchelvan and Lua, 1991)

Theory of Bulge Forming

In the bulging of a tube to an axisymmetric shape, the stress components are (Thiruvandchelvan and Lua, 1991) the hoop stress, (σ_a), the meridional stress, (σ_θ), and the stress normal to the surface, (σ_z), which is small for thin tubes and can be neglected.

Using the Tresca criterion, the equivalent stress can be expressed as :

$$\bar{\sigma} = \sigma_\theta - \sigma_a \quad (12)$$

The equivalent strain is, similarly, given by

$$d\bar{\epsilon} = \sqrt{\frac{2}{3}(d\epsilon_\theta^2 + d\epsilon_a^2 + d\epsilon_z^2)} \quad (13)$$

For volume constancy,

$$d\epsilon_\theta + d\epsilon_a + d\epsilon_z = 0 \quad (14)$$

From the free body diagram (Figure 5), equilibrium in a direction normal to the surface would give at the crown of the bulge

$$\sigma_\theta = \frac{pr_{max}}{t} - \frac{\sigma_a r_{max}}{\rho_1} \quad (15)$$

If a frictionless die is assumed, and the bending of the tube is neglected over the die radius, equilibrium of the upper part of the tube in the axial direction gives

$$\sigma_a = \frac{p(r_{max}^2 - r_0^2)}{2tr_{max}} - \frac{F}{2\pi r_{max}t} \quad (16)$$

These relationships relate geometrical and forming parameters to stresses developed in the deforming material. For bulging a tubular blank to a required shape these relationships give the stresses and strains developed in the material. The equivalent stress can be expressed as

$$\bar{\sigma} = \frac{pr_{max}}{2t} \left[1 - \frac{1}{\rho_1} \right] + \left[\frac{pr_0^2}{2t} + \frac{F}{2\pi t} \right] \left[\frac{1}{\rho_1} + \frac{1}{r_{max}} \right] \quad (17)$$

Equation 17 can also be expressed as

$$\bar{\sigma} = f[F, p, t, r_0, r_{max}] \quad (18)$$

For a given tubular blank r_0 and t are fixed and the equivalent stress can be said to depend on bulging pressure, force and

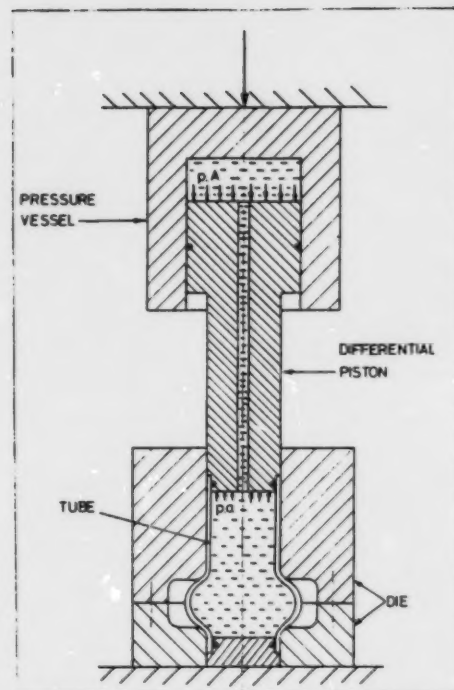


Figure 4: Schematic View Of Bulge-Forming Operation

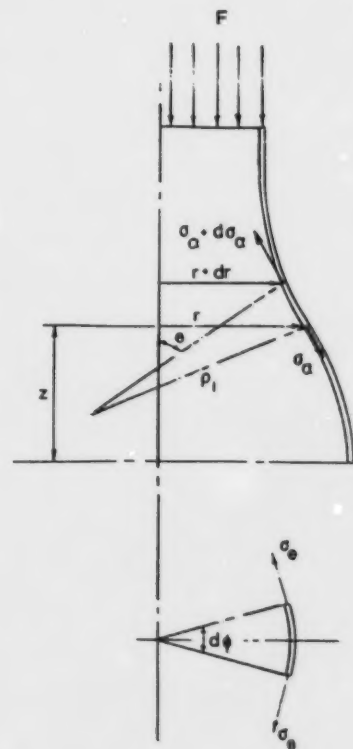


Figure 5: Free-Body Diagram

the final radius of the bulged section. The bulging pressure and force are dictated by the desired final geometry. The stresses developed in the deforming material should not exceed the failure stresses at any point during forming. The failure stress given by Equation 9 can also be expressed as

$$\sigma_f = f[i, Q, T] \quad (19)$$

For a desired final radius the value of strain-rate and temperature can be so adjusted that the maximum equivalent stress developed during bulge-forming is always lower than the failure stresses. At these values of temperature and strain-rate the tubular blank can be expected to deform without undergoing failure.

CONCLUSION

In the present paper a failure criterion for a superplastic ceramic, 20% Al_2O_3 -YTZ, has been formulated. It has also been shown how the failure criterion can be utilized to select proper temperature and strain-rate for the bulge-forming operation. A similar approach can be applied to any forming operation and optimal values of temperature and strain-rate determined.

The determination of failure stresses, however, becomes difficult due to the following reasons :

1. The constants in the Power-Law, B and m, depend on material properties, temperature and the strain-rate of deformation. An explicit relationship needs to be developed to describe these constants as a function of microstructure, temperature and strain-rate.
2. The activation energy, Q, has also been found to depend on material properties and temperature. An explicit relationship should be developed to express Q as a function of these factors.
3. The value of other constants in the failure stress expression differ from material to material. These constants should be known for the deforming material.

The developed criterion does not take cavitation effects into account. A detailed experimental study is desired to determine

the effect of strain-rate and temperature on cavitation which has been observed during superplastic deformation of ceramics. Cavitation has been found to depend on factors like strain-rate and temperature. If an explicit relationship between critical stress for the onset of cavitation and these factors is known these factors can be adjusted to minimize the effect of cavitation on failure.

Superplastic ceramics offer a great promise to the forming industry due to their ability to show unusually large deformations. However, due to the low deformation rates, the process is only economically attractive for low-volume production. Nevertheless, substantial cost and weight savings can result when a single SPF panel replaces a multicomponent assembly. Additional savings result from part consolidation due to less capital equipment required. Tool costs are dramatically reduced because matched, hard tools are not required.

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A MODEL FOR THE ANALYSIS OF POST-EXPOSURE CREEP RUPTURE DATA
OF SOLID SOLUTION STRENGTHENED ALLOYS

by

Joseph K. L. Lai
Department of Applied Science
City Polytechnic of Hong Kong
Tat Chee Avenue
Kowloon
HONG KONG

ABSTRACT

Post exposure testing involves taking samples of the service exposed material and testing under accelerated conditions. It is necessary to apply a model to obtain an estimate of the remaining life from the accelerated test data.

Purely empirical models such as the extrapolation method and the life fraction rule are currently being used. The former method requires a large number of tests and cannot be used when only a small amount of sample is available.

A semi-empirical model to analyse post exposure creep rupture data for remaining life assessment purposes is proposed. The main advantage of the present proposed model is that a microstructural damage process can be related to macroscopic creep rupture properties. By this method, a theoretical framework is provided to link the continuum damage mechanics approach to microstructural degradation mechanisms in creep rupture in a simple manner.

1. INTRODUCTION

In engineering design, it is recognised that machines and components will not last indefinitely due to various mechanisms of material deterioration such as corrosion, creep, fatigue, friction and wear etc. In one sense, a "perfect" design would be one in which the entire machine would fail completely at a given preselected life. In practice, most components will last beyond the design life time whereas a few will fail prematurely. The purpose of remaining life assessment is to identify those components which are likely to fail prematurely, and to enable the engineer to make informed decisions on whether to repair, replace or continue operation of the equipment concerned.

Research activity in the area of remaining life assessment has increased substantially in recent years. Large power generating and petrochemical industries throughout the world have recognized that plants built in the "boom" years of the 1960's and 1970's are now reaching the end of their design lives. On the other hand, the building of new plants has become increasingly difficult to justify on both political and environmental grounds. The extension of the life of existing plants by a gradual replacement of critical components is an attractive option provided that reliable life prediction methods are available.

The current remaining life assessment techniques can be divided into two categories: (1) those relying on a knowledge of the operating history, and (2) those

requiring access to the component for examination, sampling or measurement. In the first category, knowledge of the operating parameters such as temperature and stress are necessary. In the second category, current techniques include on-site plastic replication, specimen sampling for metallography and/or post exposure testing, and strain measurement using strain gauges, bow gauge micrometers or photogrammetry.

The field of remaining life assessment includes a wide range of techniques. This paper is only concerned with one aspect: the analysis of post exposure creep rupture testing data. A simple model is proposed which is particularly applicable to solid solution strengthened alloys.

2. POST EXPOSURE CREEP RUPTURE TESTING - THE NEED FOR A MODEL

Post exposure creep rupture testing involves taking samples of the service exposed material and testing under accelerated conditions (σ_A, T_A), where σ_A is the accelerated applied stress and T_A is the accelerated temperature. The methods for sample removal have been reviewed by Barraclough [1]. When only a small amount of sample is available, a "miniature" specimen technique, involving the use of electron beam welding to add extension blocks to the sample, can be used for post exposure creep testing.

Having obtained the post exposure creep rupture data at the accelerated conditions, it is necessary to apply a model to derive the estimated life from the post exposure test results. In fact, a model is inevitable unless the tests are performed under the service conditions, which are of no value for life prediction purposes. At present, there are two commonly used models - the extrapolation model, and the life fraction rule model.

The extrapolation model involves the extrapolation of short term data, either accelerated in stress or accelerated in temperature, or both, to service conditions. A variety of extrapolation methods, including parametric techniques such as Larson-Miller, Manson-Succop, Dorn, Manson-Haferd etc., may be used. This method has received general acceptance because data extrapolation is a method on which most design data are based.

The disadvantage of the extrapolation method lies in the number of data required for reliable extrapolation. In many situations, the amount of material which can be removed from the component is so small that it is only sufficient for one or two "miniature" specimens. In these situations, an estimate of the remaining life can still be made using the life fraction rule model:

$$\frac{t_1}{t_r} + \frac{t_2}{t_v} = 1 \quad (1)$$

where t_1 is the service exposed time, t_r is the creep rupture life under the service conditions, t_2 is the creep rupture life of the service exposed material tested under accelerated conditions, and t_v is the creep rupture life of the virgin material tested under the same accelerated conditions.

The use of the life fraction rule model requires a knowledge of the creep rupture strength of the virgin material. If this is not available, a reasonable estimate can be made using ISO mean properties. The scatter band of ISO data can also be used to give upper and lower bound estimates.

The problem with the life fraction rule model is that its validity is in doubt for a number of materials [2]. Moreover, both the extrapolation model and the life fraction rule model are purely empirical and are not related to any microstructural processes.

3. THE DEVELOPMENT OF A SEMI-EMPIRICAL MODEL

In this paper we set out to develop a semi-empirical model suitable for analysing post exposure creep rupture data. The requirements of the model are:

- (1) It should be simple and contain a minimum of unknown parameters for ease of application.
- (2) It should be semi-empirical in that it is possible to link it to microstructural damage mechanisms.

We begin by referring to the Kachanov's model in continuum damage mechanics [3]:

$$\dot{\epsilon} = \frac{A\sigma^m}{(1-\omega)^n} \quad (2)$$

where $\dot{\epsilon}$ is the creep strain rate, σ is the applied stress, A and m are constants, and ω is a state variable introduced to account for "damage". When this equation was introduced it was not intended to assign a microstructural meaning to this damage variable.

We now proceed to develop this purely empirical model into a semi-empirical one, linking the model to microstructural damage mechanisms. In this paper, we shall concentrate on solid solution strengthened alloys where the dominant mechanism of the loss of creep strength is solute depletion from the matrix.

Apart from macroscopic effects such as necking during a creep test, creep damage is usually attributed to the development of cavities and/or microstructural instability. Comparatively few investigations have been attempted to assess the effect of microstructural stability on creep rupture life. Instead, considerable research efforts have been devoted to study the cavity growth

aspect of creep damage.

When a creep test is performed on a solid solution strengthened alloy which has been exposed for long periods to elevated temperatures at zero or very low stresses (i.e. under most design conditions), it is often found that there is a considerable reduction in creep strength compared to the unaged or virgin material. Damage in this case is not caused by the formation of cavities, but by the depletion of solid solution strengthening elements in the region surrounding precipitate particles which are enriched in these elements. The "damage" is heterogeneous, and at any time t , we may, as an approximation, divide the total volume, V_0 , into a damaged volume, V_d , where solute elements have been depleted, and an undamaged volume, V_u , where no solute depletion has occurred.

As precipitate particles nucleate and grow, so will the "damaged volumes" of solute depleted regions. Let us assume that the x -dimension of a damaged volume nucleated at a time t' ($t > t'$) can be expressed in the form

$$x = U_x(t-t')^r \quad (3)$$

where U_x is the growth rate in the x direction and r is a parameter dependent on the details of the growth mechanism. If similar expressions are written for the y and z directions, and assuming random nucleation, it can be shown that the volume fraction of the damaged material will grow as [4]:

$$\frac{V_d}{V_0} = 1 - \exp\{-(t/\tau)^n\} \quad (4)$$

where n and τ are parameters, which are, in general, dependent on the material, the applied stress and the temperature. It is interesting to note that τ has a finite value even when the applied stress is zero. This is a special feature of this model which recognizes the fact that even stress free ageing can cause reduction in creep strength.

Let us imagine that we are performing a creep rupture test on a material which has been exposed to high temperatures at low stresses and possesses a volume fraction V_d/V_0 of solute depleted regions. The creep rupture test is to be carried out at a stress σ_0 , which is significantly greater than the stress under which the material has been exposed.

If σ_u and σ_d are the average effective stresses experienced by the undepleted and depleted regions respectively, we have

$$\sigma_0 V_0 = \sigma_u (V_0 - V_d) + \sigma_d V_d \quad (5)$$

Equation (5) assumes that the depleted regions occur randomly throughout the material and therefore at any cross section the area fraction of depleted regions is equal to the volume fraction.

Combining equations (4) and (5), we obtain

$$\frac{\sigma_u - \sigma_d}{\sigma_0 - \sigma_d} = \exp\left(\frac{t}{\tau}\right)^n \quad (6)$$

The upper bound for σ_d is determined by the strength of the depleted material. In the case when $\sigma_0 > \sigma_d$, which would be justified if solute strengthening is the dominant strengthening mechanism, we have (since $\sigma_u > \sigma_0$)

$$\frac{\sigma_u}{\sigma_0} = \exp\left(\frac{t}{\tau}\right)^n \quad (7)$$

Thus the effective stress which the undepleted material sees depends on the time of exposure in an exponential manner.

So far we have considered solute depletion as the only mechanism of "damage". If a creep rupture test is performed on a solid solution strengthened alloy at an initial stress low enough for the solute atoms to "move along" with the dislocations, according to this model, the effective stress on the undepleted regions will eventually increase to a point when the solid solution strengthening mechanism can no longer operate. As a result, a different mechanism will take over. Ultimately, cracking, cavitation or shear fracture will take place towards the end of the rupture life.

Even if the effective stress on the undepleted regions does not reach the point for a change in mechanism to occur, the fact that it increases exponentially with time means that after a time $t = \tau$, the rate of increase is so rapid that rupture must be imminent. Thus, for a creep rupture test at a low stress and a high temperature, τ should be quite close to the rupture life of the test.

We now set ω equal to V_d/V_0 in equation (2). Using equation (4), this is reduced to:

$$\dot{\epsilon} = A\sigma^n \exp\{m(t/\tau)^n\} \quad (8)$$

The initial creep rate, $\dot{\epsilon}(0)$, at $t=0$, is given by

$$\dot{\epsilon}(0) = A\sigma^n \quad (9)$$

From equations (8) and (9), we have

$$\frac{\dot{\epsilon}(t)}{\dot{\epsilon}(0)} = \exp m(t/\tau)^n \quad (10)$$

Equation (10) determines the shape of the creep curve. The parameter n can be taken as the stress exponent of the material and τ can be taken as the creep rupture life. This equation can be used to "fit" curves to creep strain data. The only variable parameter is n .

A numerical integration method (area under curve) is used to generate creep curves (i.e. ϵ vs t) for various values of n . This method is used to "fit" curves to the long term creep strain data of a cast of AISI type 316 stainless steel. The creep test was conducted at 600°C at a stress of 115.8MPa and was part of a programme of long term creep tests reported previously [5]. The rupture life was 98769h. Fig.1 shows the set of creep curves generated for various values of n . The actual creep strain data are also plotted in the figure for comparison.

It is necessary to assign a stress exponent for this material at this test condition. A survey of available data [6] shows that the creep stress exponent for AISI type 316 stainless steel varies with stress and temperature. At high stresses the value of n can be quite high and is of the order of about 10. At lower stresses the value of n decreases, and a value of 4 is appropriate for the test condition indicated above. τ is taken to equal to the rupture life of 98769h. $\epsilon(0)$ is obtained from the initial creep strain data. It can be seen in Fig.1 that, with the above mentioned values for m , τ and $\epsilon(0)$, $n = 2$ gives a good fit to the creep strain data. In fact, the actual value of n for the best fit is slightly higher than 2.

4. ANALYSIS OF POST-EXPOSURE CREEP RUPTURE DATA

We now proceed to apply the model developed above to analyse post-exposure creep rupture data. Post-exposure testing involves taking samples of a pre-crept (or service exposed) material and testing under accelerated conditions of stress and/or temperature.

Let σ_v be the (accelerated) applied stress required to rupture the virgin material at a time t_r , and σ_e be the corresponding applied stress required to rupture the pre-crept material at the same rupture time, t_r , and at the same temperature. Although the applied stress on the pre-crept material is σ_e , the undepleted regions in this material is in effect seeing an applied stress of σ_v because the rupture time, t_r , is the same as that of the virgin material at the stress σ_v . The reduction from σ_v to σ_e is, of course, due to the fact that a volume fraction, V_d/V_0 , was "damaged", for example, by the depletion of strengthening elements, during prior exposure.

From equation (7), we obtain

$$\frac{\sigma_v}{\sigma_e} = \exp \left(\frac{t_r}{\tau} \right)^n \quad (11)$$

where we have replaced the effective stress, σ_e , by σ_v , and the applied stress, σ_e , by σ_e ; t_r is the time of exposure to prior creep.

The ratio σ_v / σ_e can be obtained by creep rupture testing of the virgin and post-exposed materials. As in the case of the life fraction rule model, if the virgin material properties are not available, ISO data may be used to give the mean and the upper and lower bound limits. Since t_r is known, τ can be calculated from equation (11) provided n is known. As mentioned previously, τ is quite close to the rupture life of the material at the pre-creep (or service exposure) conditions. This allows an estimate of the remaining creep life to be made.

The value of n can be derived from microstructural considerations of the solute depletion mechanism [7], which gives a value of n equal to 2.5. Thus, unlike purely empirical models, where all unknown parameters have to be determined empirically, this semi-empirical model allows one of the critical parameters to be determined by theoretical analysis of microstructural mechanisms. The value of $n = 2.5$ was found to be in good agreement with experimental data on AISI type 316 stainless steel [7].

5. DISCUSSION

In order to analyse post exposure creep rupture data for remaining life assessment purpose a model must be used. When only a small amount of sample is available, the extrapolation model cannot be used, and the life fraction rule becomes the only model available up to now.

Like the life fraction rule model, it is possible that the present proposed model can be successful in some situations but not in others. However, the main advantage of the present proposed model is that a microstructural damage process can be related to macroscopic creep rupture properties through the parameter n . By this method, a theoretical framework is provided to link the continuum damage mechanics approach to microstructural degradation mechanisms in creep rupture in a simple manner.

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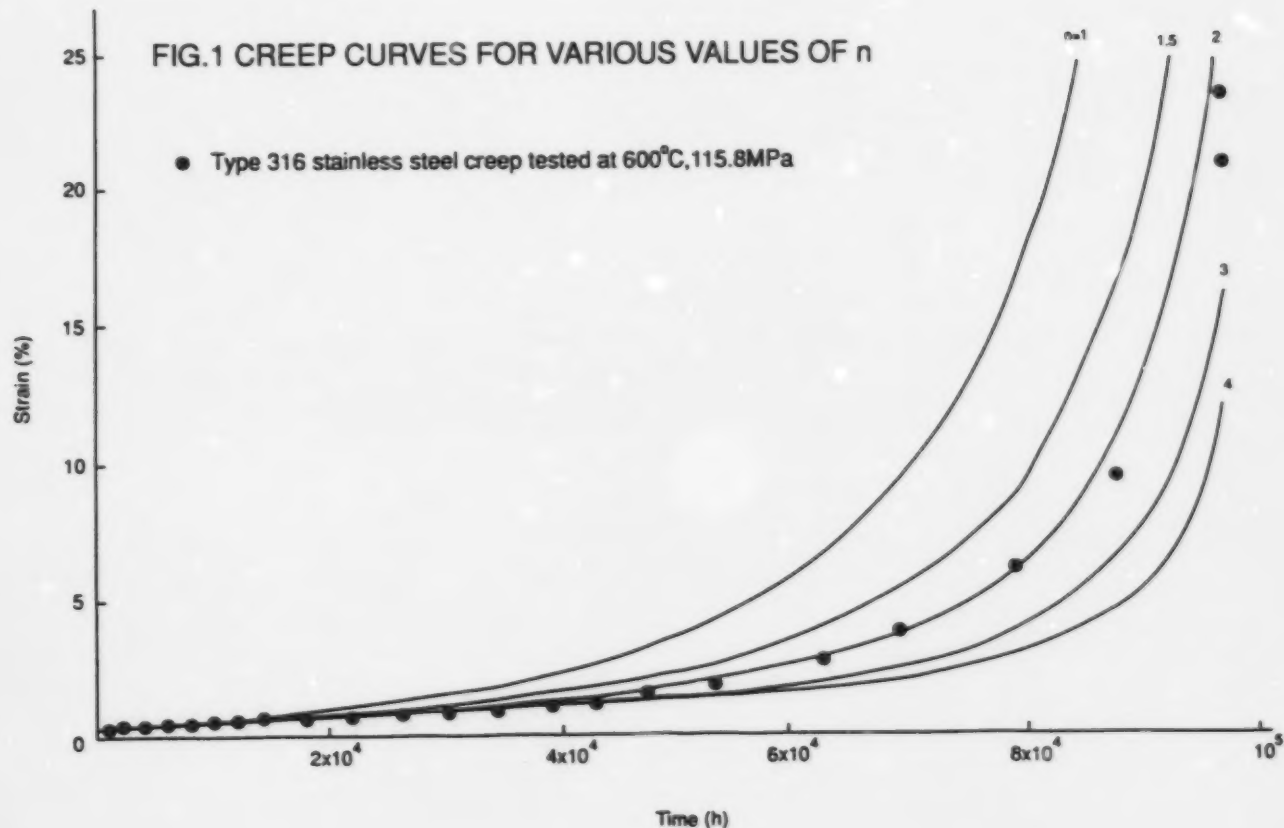
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FINITE ELEMENT MODELING OF ORTHOGONAL CUTTING

M. Jayashankar and S. D. Bhole
Industrial Systems Engineering, University of Regina,
Regina, Saskatchewan, Canada S4S 0A2.

ABSTRACT

A preliminary investigation has been made to model the Orthogonal Cutting Process using a Finite Element Analysis general purpose computer software package ANSYS. A careful selection of element types and boundary conditions has been made so as to reflect the actual components and operating conditions. Two models have been developed, one depicting the "start of the cutting operation" and the other one characterizing the "cutting in progress". In these models, the work piece is made stationary and the tool is allowed to move in a horizontal direction with the force applied to the tool. Both the models predict that the deformation starts very close to the tool tip and moves ahead of the tool as the tool is fed in. The maximum shear stress occurs at those elements which collectively form a deformation zone, where the shear stress reaches its maximum value confirming the start of yielding. It was found that the deformation zone or the "shear zone" is inclined at an angle to the direction of relative movement of the tool-work piece. Further, it was noted that the shear zone is not a plane but has substantial width. The above findings agree with the accepted theories of Merchant and Oxley and thus the finite element models may be used for future analysis and research.

INTRODUCTION

All metal cutting operations involve removing the unwanted metal away from a metal piece by the use of a tool of harder material, to make a final product of the required shape. The tool is wedge-shaped and has a straight cutting edge. It is constrained to move relative to the metal piece, usually termed as "work piece", in such a way that a layer of unwanted metal is removed in the form of a chip. Although a considerable amount of effort has been expended in the analysis of the metal cutting problem, the mechanics of the chip formation process are still far from clear. Several solutions have been proposed to the metal cutting problem by considering the orthogonal cutting (explained in the next section of the paper), to be a deformation in plane strain. Ernst and Merchant[1] based their model (Figure 1) on the minimization of work done in cutting, which resulted in the so-called "shear angle" solution. Lee and Shaffer[2] proposed an idealised shear plane model (Figure 2) where all the deformation takes place in a plane extending from the tool cutting edge to the point of intersection of the free surfaces of the work and chip, resulting in a triangular plastic zone. Palmer and Ox-

ley[3] predicted (Figure 3) that the deformation zone is not a plane but has substantial width with the stream lines of flow following smooth well defined curves whose curvature increased towards the cutting edge. Ramalingam[4] by his experiments questioned the assumption of deformation in plane strain. The influence of microcracks on the deformation of metal while cutting was analysed by Luong[5]. All the above theories, analysis and experiments tried to find out the location, dimensions and configuration of the deformation zone and the mode of its formation. With the availability of high speed digital computers and with the growing emphasis on numerical methods of engineering analysis, newer methods have been developed and made available to the engineers in the form of highly powerful computer software. One such method is the Finite Element Analysis used for problems involving complex material properties and boundary conditions. An attempt has been made here to develop a finite element model of orthogonal metal cutting and to validate it using existing theories to find out whether the developed model can be utilized for further analysis and research.

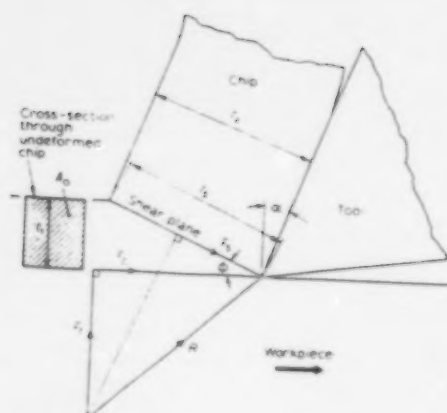


Figure 1: Model of Ernst and Merchant.

ORTHOGONAL CUTTING

Orthogonal cutting is a special case of the cutting operation. Here the straight cutting edge of the tool is perpendicular to the direction of relative motion of the work piece and the tool as shown in Figure 4. The main advantage of this arrangement is that it represents a two dimensional rather than a three dimensional problem and is quite useful in theoretical and experimental work as many of the independent variables are eliminated. However, this arrangement is seldom used in practice. Oblique Cutting as shown in Figure 5 is the arrangement used in almost all practical metal cutting operations. In oblique cutting, the cutting edge of the tool is at an angle to the relative work piece-tool movement. The advantage of this arrangement is that the flow of the chip while cutting, would be in a direction away from the tool creating less tool wear.

DEVELOPMENT OF FINITE ELEMENT MODEL

The various steps involved in the development of the Finite Element model are:

- (i) Selection of suitable element types for the tool and work piece.
- (ii) Identification of boundary conditions
- (iii) Development of the tool and work piece model using the commercially available computer package ANSYS[6].
- (iv) Identification of the stress zones
- (v) Comparison of the location of stress zones with previous findings

Following the above steps, a complete finite element model of orthogonal cutting has been developed as detailed below.

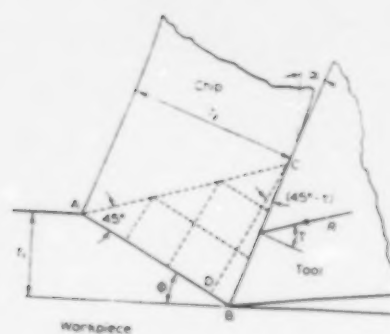


Figure 2: Model of Lee and Shaffer.

Selection of Element Type

A successful finite element modeling starts with employing appropriate element types for the components to be modeled. These elements should be so selected that their behavioral pattern matches with that of the material during the actual conditions. A variety of elements are available to suit different analysis and materials.

In orthogonal cutting, the components involved are the work piece and the tool. Both of them would be of similar material except that their characteristics like toughness, stiffness etc., would be different.

A two dimensional solid element has been selected for modeling the work piece. The element is defined by four nodal points having two degrees of freedom at each node with translations in the nodal x and y directions. This element type is selected for the following reasons:

- (i) It can be used in the modeling of solid structures.
- (ii) It can be used for plane strain problems. Since orthogonal cutting is being analysed as a plane strain problem, this type of element suits the purpose.
- (iii) The element has plastic, stress stiffening and large deflection capabilities which are encountered in a cutting operation.
- (iv) The element has a three nodal option, i.e., it can be used as a triangular element with three vertices.

This element is described as STIF42 in the ANSYS computer package. The same is used for modeling the tool also because of its capability to accept the three nodal description (see iv above).

A second element type which is vital for this modeling has also been employed. This is a two dimensional interface element representing two surfaces which may maintain or break physical contact. The element has two degrees of freedom at each node with translations in the nodal x and y directions. This element

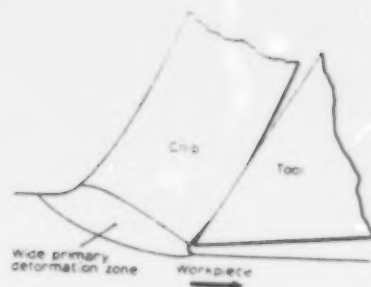


Figure 3: Model of Palmer and Oxley.

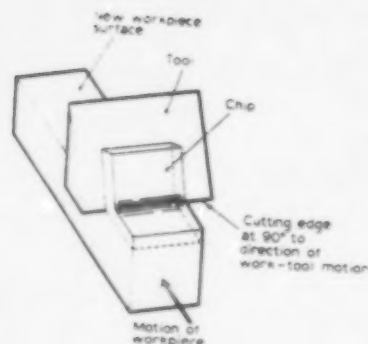


Figure 4: Orthogonal Cutting.

is used at the point where the tool makes contact with the work piece. It is also used at locations where the chip comes into contact with the tool face. This element type is selected for the reasons given below:

- (i) It is a two dimensional element.
- (ii) It represents two surfaces which may maintain physical contact or break according to the conditions developed. Since metal has to part away during cutting, this characteristic of the element suits the requirement.
- (iii) It can slide relative to its other surface. As the chip has to slide over the tool during cutting, this capability of the element serves the purpose.
- (iv) The element is capable of supporting compression in the direction normal to the surfaces and shear (coulomb friction) in the tangential direction. As the metal is getting sheared away during cutting, the capacity of this element to support the force would allow the generation of respective stresses resembling actual cutting conditions.

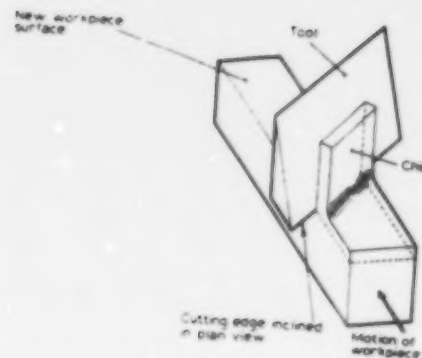


Figure 5: Oblique Cutting.

Identification of Boundary Conditions

Boundary conditions are normally the constraints to be imposed on the model so that it will resemble the actual performance of the component at the start of and during the actual cutting.

Since there are two components, namely tool and the work piece, both have to be given proper boundary conditions. In this model, the work piece is made stationary and the tool is applied with a specified cutting force to cut the work piece. The work piece is pinned at the bottom nodes so that its translational movements in the x and y directions are restricted. The tool is pinned at its nodal points so that it is constrained to move only in the horizontal x direction.

Finite Element Model

As mentioned in the introduction, the modeling can be done using computer software tailor-made to perform finite element analysis. One such software is "ANSYS" which is used in this project for modeling, computation and interpretation of results called post processing.

Two models have been developed; one depicting the start of the cutting operation named Model-1 and the other showing "cutting in progress" called Model-2.

Model-1 (Figure 6) consists of a work piece 100mm long (x direction) and 50mm high (y direction). Using the 4-node element STIF42, the work piece has been discretized. At the point where the tool touches the work piece, an interface element STIF12 has been incorporated for the material to part-away from the work piece. The tool has been developed with the same STIF42 element using its 3-nodal option. A 40° rake angle and a 5° clearance angle have been incorporated on the tool.

Model-2 (Figure 7) is quite different from Model-1 in that it represents one instance of "cutting in progress" after the tool has travelled 50mm along the x direction. Here the tool is discretized into eleven parts using 4-nodal element. This is done to connect the chip to the tool by the interface element STIF12 so that the chip can slide over the tool rake face. The work piece is modeled using the STIF42 elements as before.

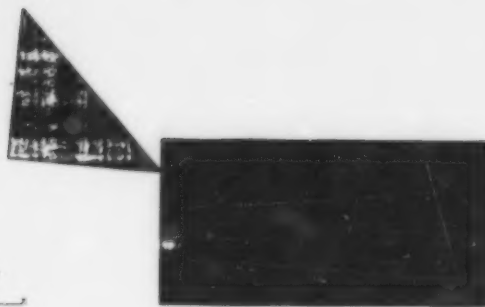


Figure 6: Model-1.



Figure 7: Model-2.

The properties of the work piece and tool material considered in this paper are the same for both the models and are listed in the table below:

Component	Young's Modulus	Yield Stress
Work piece	2425.6 N/mm ²	63.3 N/mm ²
Tool	10x10 ¹⁰ N/mm ²	-

Since the stresses developed in the tool during the cutting operation are not critical to this study, it is made highly stiff using a high Young's modulus value compared to the work piece.

A value of 0.5 is given as coefficient of friction and a value of 2.4256×10^5 N/mm² is assumed as tangential stiffness for the interface element incorporated between the tool tip and the work piece contact node. Frictionless condition and zero tangential stiffness are assumed for the interface elements incorporated between the tool rake face and the chip.

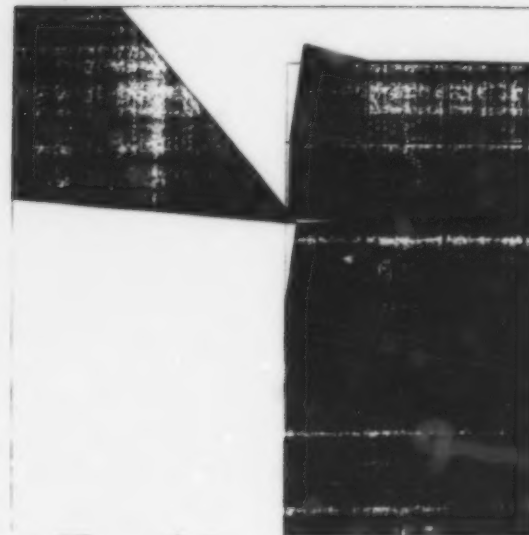


Figure 8: Displacement Plot of Model-1.

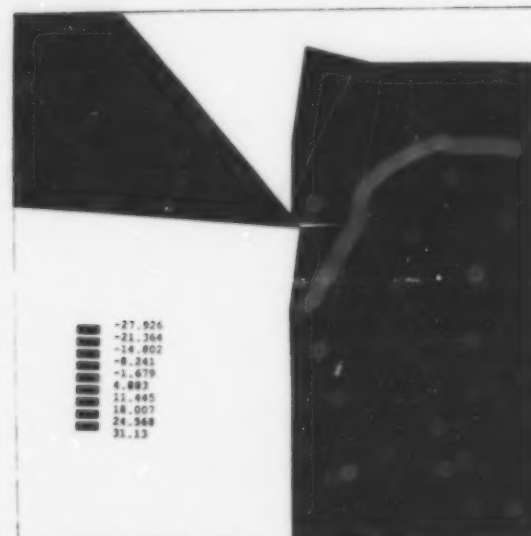


Figure 9: Shear Stress Plot of Model-1.

RESULTS

Model-1

A force of 15N has been applied in the x direction on the nodal points of the tool and finite element analysis has been carried out. The node at which the tool contacts the work piece has been displaced by a distance of 0.1234mm and the maximum displacement occurring here can be seen from the displacement plot in Figure 8. The element above the contact point is parting away and the displacement of this element is in the negative y direction. A force applied to the top right corner node of the work piece makes the chip to move upwards.

Referring to Figure 9, which is a plot of shear stress, the stress is maximum on those elements which are closer to the tool tip and is in the range of 24 - 31 N/mm² on the element

just above the tool tip. In all other elements, it is comparatively lower. The results also show that the combined or equivalent stress is higher on those elements nearer to the tool tip than that in any other element in the work piece. This stress is in the range of 60 to 69 N/mm².

Model-2

As this model depicts the stage of cutting in progress, the same load of 15N has been applied in the x direction on the tool and the finite element analysis has been carried out. The contact nodal point between the tool and the work piece is displaced by a distance 0.03052mm in the x direction and the maximum displacement occurs at this node. The tool and the chip above the tool have moved in the x direction. The displacement of the tool is comparatively larger than that of the chip. The shear stress is found to be maximum on certain elements near the tool tip. Referring to Figure 10 the elements above the tool tip along the direction of x axis are stressed more than any other element in the work piece. This shows that deformation occurs in this zone prior to any other zone in the work piece.

FINDINGS AND DISCUSSION

The present results should be compared with known results from previous theoretical studies and/or from experiments conducted earlier to check its validity and reliability. Referring to Model-1, which shows the effect at the start of cutting, the deformation zone in the work piece starts forming very close to the tool tip and as the tool moves, it deforms the metal in front of the tool tip. The shear stress reaches its maximum value confirming the start of yielding of the work material and a gradual parting away of the chip can be noticed in the plot (Figure 9). However, there is no tendency for the chip to move up the tool rake face. This is due to the fact that there is no acceleration of the work piece or tool to make the chip move up the tool. Under actual cutting conditions, the tool or the work piece is made to move at a certain velocity which provides the necessary momentum for the chip to move up. Hence in the present model, a force is applied to the top corner element of the work piece which may be equivalent to the acceleration, to make the chip move upwards.

Considering Model-2, it was observed that the displacement of the element in the work piece very near to the tool tip is lower than that found in Model-1, probably because the machined material, in other words, extra elements attached to them, provide the additional stiffness against displacement. Further, the tool has moved in the horizontal x direction more than the chip resting on the tool rake face. This implies that the chip really moves up, which happens during the actual cutting process. Indeed, a negligible amount of force is required to maintain the stability of the chip.

A scrutiny of the stress values shows that the maximum shear stress occurs at those elements which form a stress zone or deformation zone. The configuration of the deformation zone shows that it has substantial width and an angle of inclination to the horizontal, which is the "shear angle" as mentioned by Ernst and Merchant[1]. Since the stress on these elements is more than any other element in the work piece, it may be deduced that the deformation occurs here and the metal becomes plastic to part away from the work piece in the form of a chip. These findings

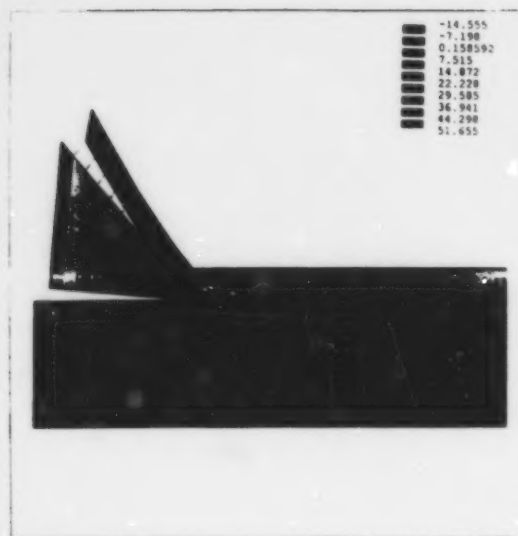


Figure 10: Shear Stress Plot of Model-2.

are in good agreement with the existing theoretical results as listed below:

- (i) There exists a deformation zone or shear zone in orthogonal cutting, where the metal deforms and parts away from the work piece, as predicted by all the earlier theories.
- (ii) The deformation zone is not a plane but has substantial width as shown by Palmer and Oxley[3].
- (iii) The plastic deformation starts when the stress reaches a value of maximum shear stress.

Thus, the developed models are satisfactory for utilization in future analysis and research.

CONCLUSION

A preliminary study has been made to show that a large deformation process like metal cutting can be modeled and analysed using a general purpose finite element program ANSYS. Results of analysis using the model were in good agreement with earlier work regarding deformation zones and cutting forces. The model can be used for evaluating the effects of rake angle, tool tip radius and stresses developed in the tool. It can also be used to understand the chip formation, temperature effects, etc. Further analysis and usage of this model will definitely throw more light on the mechanism of Orthogonal Cutting.

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TECHNICAL SESSION 3

MANUFACTURING

Chair: M. Chen



AN OPTIMAL TOOL REPLACEMENT MODEL

M. Liang and S. Taboun

Department of Mechanical Engineering
University of Ottawa
Ottawa, Ontario, Canada K1N 6N5

ABSTRACT

In this paper, a new model is proposed to determine the optimal tool replacement time with consideration of both production loss cost and in-process failure penalty. The proposed model is valid for general tool life distribution. An explicit expression of optimal tool replacement age has been developed for Weibull distribution. An illustrative example is included.

INTRODUCTION

Tooling problem has been recognized as a critical issue in manufacturing, especially in automatic manufacturing (Gray et al 1988, Tomek 1986, Elmaraghy 1985). An important goal of tool management is to achieve optimal tool utilization, which is directly influenced by tool replacement decision. The commonly used tool replacement strategies are

- (a) scheduled tool replacement;
- (b) preventive planned tool replacement;
- and
- (c) failure tool replacement.

In the scheduled tool replacement strategy, tool is replaced either at a specified time or upon failure, whichever occurs first. In the second strategy, a tool is replaced when a specified number of workpieces has been processed, or upon failure. The last strategy is to replace the tool only when it has failed.

In literature, the tool replacement problem has been studied by a number of researchers. The earlier research include the work by Doyle (1973), and the work by Kendall and Sheik (1979). The methods suggested in their papers did not consider the fact that the workpieces must be rejected upon an in-process tool failure in the automated manufacturing. La Commare et al (1983) developed three models associated with the

three tool replacement strategies mentioned above. To take into account in-process failure, a penalty cost is included in their models. Their study shows that the scheduled replacement strategies is superior as compared to the other two strategies. Their work was later extended to incorporate the effect of preliminary running-in of the cutting-edge by Noto La Diega and Passannati (1984). Zhou et al (1990) proposed a model based on the actual tool wear status. In their model the probability distribution of the tool capacity and the tool capacity consumption by the workpieces are separately considered. The objective is to find the optimal replacement point in terms of residual tool capacity such that the tool utilization and in-process tool failure are balanced.

It is noticed that

a) the production loss cost due to tool replacement has not been considered in previous studies. More frequent tool replacement may, on one hand, achieve a better trade-off between tool utilization and in-process failure, but on other hand, lead to a higher production loss due to the increased downtime.

b) an explicit expression of the optimal tool replacement point is not available yet. Based on the previous models, the optimal tool replacement point is generally obtained by direct search or solved numerically. To further exploit the relationships between various factors and the optimal replacement

point, an explicit expression of the optimal replacement point is obviously desirable.

The main purpose of this research is, therefore, to develop an optimal tool replacement model which takes into account both in-process penalty cost and production loss cost due to the tool replacement, and to induce an explicit expression of the tool replacement point. In view that the tool wear status may not be available in many manufacturing systems and that the scheduled replacement strategy has shown to be superior to both preventive and failure replacement strategies (La Commare et al 1983), we will develop a new model based on the scheduled replacement strategy.

MODEL FORMULATION

In most previous scheduled replacement based models (e.g., La Commare et al 1983, and Noto La Diega and Passannati 1984), time was treated as an integer multiple value (number of workpieces) of the machining time of a single workpiece. As a result, if the ratio of tool life to machining time needed for a single workpiece is small, say, less than 1, the models may not be valid or cause low tool utilization. To avoid this problem, the tool replacement time will be treated as a continual variable in this paper.

This model develops optimal scheduled replacement policy for a single tool with a time-to-failure distribution characterized by an increasing failure rate. The tool is replaced whenever a failure is detected or upon a specified tool age. While replacing a tool, the machine is down, leading to production loss. In this paper the production loss cost is assumed to be linearly related to replacement time. In addition, when there is an in-process tool failure, the penalty cost occurs. It includes the cost of scrapped workpiece or the extra time required to re-machine the defective workpiece, additional machine adjustment, etc. If a tool is replaced too early, the remaining tool capacity will be lost. Frequently tool replacement also costs excessive machine down time. On the other hand, if a tool is replaced too late, the probability of tool failure in-process can be very high. The optimal scheduled replacement policy is, therefore, to balance tool utilization, in-process failure, replacement cost and production loss cost. The goal of this model is to determine when a tool should be replaced to minimize the cost. The total production cost is expressed as a function of the replacement age. In order to determine the optimal replacement age, the total production cost is minimized with respect to the replacement interval. The model will be formulated based on the following notations.

- C_N the total cost per cycle
- C_o the machining cost per unit time, including labour cost, overhead and other time dependent costs
- C_p production loss cost per unit time

- C_{rs} the cost of replacing tool before failure, including the cost of a new tool
- C_{ru} the cost of replacing tool after failure, including the cost of a new tool
- C_{pe} penalty cost, resulting from an in-process failure
- N the age at which scheduled replacement is performed
- $R(N)$ the tool reliability function
- t_{rs} the time spent in the scheduled tool replacement
- t_{ru} the time spent in the unscheduled tool replacement

The expected total production cost over a single cycle consists of operation cost, tool replacement cost, production loss cost and penalty cost. Denoting expected total production cost by $E(C_N)$ yields:

$$E(C_N) = C_o E(T) + C_{rs} R(N) + [C_{ru} + C_{pe}] [1 - R(N)] + C_p [t_{rs} R(N) + t_{ru} (1 - R(N))]$$

or

$$E(C_N) = C_o E(T) + [C_{rs} + C_p t_{rs}] R(N) + [C_{ru} + C_{pe} + C_p t_{ru}] [1 - R(N)] \quad (1)$$

The expected cycle length $E(T)$, including the time of replacement and the time spent in production, can be written as:

$$E(T) = U_N + t_{rs} R(N) + t_{ru} [1 - R(N)] \quad (2)$$

where U_N is the expected good time over the cycle:

$$U_N = \int_0^N R(t) dt \quad (3)$$

Then the expected total production cost over the cycle is expressed as follows:

$$E(C_N) = C_o [U_N + t_{rs} R(N) + t_{ru} (1 - R(N))] + (C_{rs} + C_p t_{rs}) R(N) + (C_{ru} + C_{pe} + C_p t_{ru}) (1 - R(N)) \quad (4)$$

The average cost per unit good time is:

$$\frac{E(C_N)}{U_N}$$

Dividing it by C_o , a dimensionless measure $E(C_{UN})$ is obtained:

$$E(C_{UN}) = \frac{E(C_N)}{U_N C_o}$$

This dimensionless value indicates the ratio of total cost to the operating cost. Obviously, it is desirable to reduce this ratio. This is because that a lower $E(C_{UN})$ indicates lower "unnecessary" costs including undesired replacement cost, cost due to lower tool utilization, in-process failure penalty and production loss cost. When $E(C_{UN})$ approaches unity, the "unnecessary" costs approaches zero. The best replacement time will be the one corresponding to the lowest $E(C_{UN})$ value. Therefore, $E(C_{UN})$ is used as a criterion of replacement goodness in this paper. $E(C_{UN})$ can be written as:

$$E(C_{UN}) = \frac{1}{U_N} ((U_N + t_{UN} R(N) + t_{UN} (1 - R(N))) + \frac{1}{U_N C_o} [(C_{UN} + C_p t_{UN}) R(N) + (C_{UN} + C_{pu} + C_p t_{UN}) (1 - R(N))])$$

or

$$E(C_{UN}) = \frac{1}{U_N} [U_N + t_{UN} R(N) + t_{UN} (1 - R(N))] \quad (5)$$

where t_{UN} may be interpreted as the total imputed downtime for scheduled tool replacement:

$$t_{UN} = t_{UN} + \frac{C_{UN} + C_p t_{UN}}{C_o} \quad (6)$$

and t_{UN} is interpreted as the total imputed downtime for unscheduled tool replacement:

$$t_{UN} = t_{UN} + \frac{C_{UN} + C_{pu} + C_p t_{UN}}{C_o} \quad (7)$$

To obtain the optimal tool utilization for scheduled replacement, $E(C_{UN})$ is differentiated and the derivative is set to zero.

$$E'(C_{UN}) = \frac{1}{U_N^2} ((U_N' + t_{UN}' R'(N) + t_{UN}' (1 - R'(N))) U_N - U_N' (U_N + t_{UN} R(N) + t_{UN} (1 - R(N))))$$

Notice that

$$U_N' = \frac{d}{dt} [\int_0^N R(t) dt]_{t=N} = R(N)$$

set $E'(C_{UN}) = 0$ and divide both sides by $R(N)$, it follows:

$$[1 + t_{UN} \frac{R'(N)}{R(N)} - t_{UN} \frac{R'(N)}{R(N)}] U_N - [U_N + t_{UN} R(N) + t_{UN} (1 - R(N))] = 0$$

This gives the expression:

$$\lambda(N) \int_0^N R(t) dt - [1 - R(N)] = \frac{t_{UN}}{t_{UN} - t_{UN}} \quad (8)$$

where

$$\lambda(t) = - \frac{R'(t)}{R(t)} \quad (9)$$

Let

$$P_u = \frac{t_{UN} - t_{UN}}{t_{UN}} \quad (10)$$

P_u can be considered as the imputed penalty for an in-process tool failure. Then, equation (8) can be re-written as:

$$\lambda(N) \int_0^N R(t) dt - [1 - R(N)] = \frac{1}{P_u} \quad (11)$$

Based on equation (11), the optimal replacement interval N can be determined.

DETERMINING THE OPTIMAL REPLACEMENT AGE FOR TOOLS WITH WEIBULL DISTRIBUTION

The Weibull distribution has been recognized as one of the most used distribution functions that define the tool life stochastic nature (Noto La Diega and Passannati 1984). In this section, an explicit expression of the optimal scheduled tool replacement age for tools with Weibull distribution is induced based on equation (11).

The probability density function, $f(t)$, is,

$$f(t) = \alpha \beta t^{\alpha-1} \exp(-\beta t^\alpha) \quad (12)$$

where α and β are shape and scale parameters respectively. The reliability function for this distribution is given by:

$$R(t) = \exp(-\beta t^\alpha) \quad (13)$$

The failure rate, $\lambda(t)$, is expressed by:

$$\lambda(t) = \alpha \beta t^{\alpha-1} \quad (14)$$

Substituting equations (13) and (14) into equation (11), we get:

$$\alpha \beta N^{\alpha-1} \int_0^N \exp(-\beta t^\alpha) dt - (1 - \exp(-\beta N^\alpha)) = \frac{1}{P_u} \quad (15)$$

let

$$x = \beta N^\alpha \quad (16)$$

$$y = \beta t^\alpha \quad (17)$$

and substituting x and y into equation (15), this becomes,

$$x^{(s-1)/s} \int_0^x y^{1/s-1} \exp(-y) dy - 1 + \exp(-x) = \frac{1}{P_u} - 0$$

or

$$x^{(\alpha-1)/\alpha} \Gamma(x, 1/\alpha) - 1 + \exp(-x) - \frac{1}{P_u} = 0 \quad (18)$$

where

$$\Gamma(x, 1/\alpha) = \int_0^x y^{1/\alpha-1} \exp(-y) dy \quad (19)$$

which is an incomplete Gamma function. Rewriting equation (18), we get:

$$P_u = \frac{1}{x^{(\alpha-1)/\alpha} \Gamma(x, 1/\alpha) - 1 + \exp(-x)} \quad (20)$$

If P_u is large, x approximates asymptotic equation (Jorgenson 1967):

$$x = \frac{1}{(\alpha-1) P_u} \quad (21)$$

For small values of P_u , x approximates asymptotic equation:

$$x = \left[\frac{P_u + 1}{P_u \Gamma(1/\alpha)} \right]^{\alpha/(\alpha-1)} \quad (22)$$

where

$$\Gamma(1/\alpha) = \int_0^\infty y^{1/\alpha-1} \exp(-y) dy \quad (23)$$

which is a Gamma function and the tables for this function can be found in (Beyer 1978). From equation (16), the optimum tool replacement age is

$$N^* = \left(\frac{x}{\beta} \right)^{1/\alpha} \quad (24)$$

ILLUSTRATIVE EXAMPLE

The following data have been assumed for the tool failure characteristics that are closely represented by a Weibull distribution with β equal 1 and α equal 2. Other data are assumed as follows:

$$t_{rs} = 0.05 \text{ hr}, t_{ru} = 0.1 \text{ hr}, C_{rs} = \$10, C_{ru} = \$12, C_{pe} = \$20, C_p = \$180/\text{hr}, \text{ and } C_o = \$120/\text{hr}.$$

Using equations (6), (7), and (10), we obtain:

$$t_{sd} = 0.208, t_{ud} = 0.517, P_u = 1.484$$

Based on equation (32),

$$x = \left[\frac{1.484 + 1}{1.484 \Gamma(1/2)} \right]^2$$

Considering the relation:

$$\Gamma(n) = \frac{\Gamma(n+1)}{n}$$

it follows

$$\begin{aligned} \Gamma(1/2) &= \frac{\Gamma(1+1/2)}{1/2} \\ &= 2 \times 0.8862 \\ &= 1.7724 \end{aligned}$$

$$\begin{aligned} x &= \left(\frac{2.484}{1.484 \times 1.7724} \right)^2 \\ &= 0.892 \end{aligned}$$

substituting x , α and β into equation (24), we get

$$\begin{aligned} N^* &= \left(\frac{0.892}{1} \right)^{1/2} \\ &= 0.944 (\text{hr}) \end{aligned}$$

The optimum tool replacement period is found to be 0.944 hours.

If production loss cost is not considered, i.e., $C_p = 0$, and other data are unchanged, then, based on equation (24), the optimal replacement age will be 0.886 hours, indicating more frequent tool replacement.

If in-process failure penalty cost reduces to \$10, and other data are unchanged, the optimal replacement age will increase to \$1.085 hours.

CONCLUDING REMARKS

A new tool replacement model is proposed to seek a trade-off among: labor cost, overhead cost, replacement cost (including tool cost), in-process failure penalty and production loss cost. An useful expression of optimal replacement age for Weibull distribution has been induced from the proposed model. The effect of production loss cost on replacement decision has been demonstrated through an illustrative example.

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Grasping with Compliant Fingers: Stiffness of Power Grasps

Shahram Payandeh
Assistant Professor

Experimental Robotics Laboratory (ERL)
School of Engineering Science
Simon Fraser University
Burnaby, British Columbia
CANADA

ABSTRACT

It has been shown in the literature that if the grasp between multiple robotic mechanisms and objects has spring (compliant) property, it is a stable grasp. The grasp between the multiple mechanisms and the object can be established in variety of configurations. It can be for example a precision one, where the tips of the mechanisms make contact with the object or the power grasp, where in addition to the tips, the segments of the finger also make contact with the object. This paper first presents a formal approach for modelling the contact between the robotic mechanisms and the solid objects. Based on the assumption that both the segments of the mechanisms and the objects are solid bodies and the causal representation of the mechanical elements, a contact model is proposed. The contact is modelled as springs and dampers where they are connected in parallel between the end-points of the mechanisms and the object. The compliant properties can be obtained from the rubber-type material attached to the segments of the fingers. It is discussed that as long as the model of the compliant materials are available, the grasping forces can be regulated by robust position controller approach. Next, a class of power grasp between the mechanisms and the object is defined. Previously, a method for determining the stiffness of the precision grasp configuration has been proposed. In this paper, the earlier results of the approach for determining the stiffness are extended to the power grasp configuration. It is discussed how the stiffness of the grasped object can be change by using the geometrical nonlinearity of the compliant material.

INTRODUCTION

In application of robot manipulators to peg-in-hole tasks, it was realized that when the end-point of a manipulator is equipped with a compliant device, the held peg can be incorporate the position uncertainties during the peg insertion for successful assembly [1]. The device which referred to as Remote Center Compliance (RCC) is attached between the end-point of the manipulator and end-effector and has inherent spring and damping properties. This allows the gripping mechanisms to be displaced from its equilibrium configuration in the presence of external forces, i.e. *passive approach for introducing compliance*. A similar spring behavior of the end-effector can be achieved through position and velocity feedback gains of the manipulator end-point controller [2]. (i.e. *active approach for introducing compliance*).

In applications of multiple robotic mechanisms such as in design of dexterous end-effector, the presence of compliance plays a similar role, as in a single manipulator application. Here the spring property of the grasp can be achieved using either active or passive approaches. In the passive approach, compliance is introduced at the tip of the mechanism, either through attachment of rubber like (spring like) material [3], or some other alternative medium, which also have compliant properties [4]. The other approach is the active one, where compliance is introduced through feedback gains of the finger end-points positions and velocities [5].

It was also shown, that when the object is grasped by mechanisms which has spring properties, the force closure grasp is stable, when it is perturbed from its equilibrium. This is equivalent to the case that when inserting a grasped peg into a solid chamfered hole. The grasped peg can be displaced from its equilibrium configuration to accommodate in interaction forces and it will return to this configuration when the forces are removed.

In addition to creating a stable grasp, it was discussed by [6], that in order for the force controller of a manipulator be stable, either the manipulator, or the contacting environment must have compliant property. It was also discussed by [7] that the presence of compliant material between the rigid end-point of the manipulator and the rigid environment reduces the loop-gain of the closed loop system and allows a fast and stable response of the contact force controller. This paper presents a model of grasp between the compliant fingers and objects where in addition to the finger-tips, the segments of the fingers also make contact with the objects.

This paper is organized as follows: section (2) presents a formal approach for modelling the fingers of the grasping mechanism; section (3) defines a class of power grasp between the compliant fingers and the object and how the stiffness of the object can be changed using the compliant properties of the contacting fingers; section (4) presents methods for controlling the grasping forces as a function of the compliant materials, position of the fingers given the model of the compliant material and finally section (5) presents discussions and highlights the future research.

MODELS OF FINGER

This section presents models of grasping mechanisms in contact with an object. First, a general contact model between two rigid

bodies is described. Then models of fingers are proposed based on the general contact model between two rigid bodies. The definition of the stability of the object grasped by fingers having compliant (spring) property, will also be reviewed.

The general model of contact between grasping fingers and object is obtained based on the *Principle of Causality* and the causal representation of the mechanical elements. In general, causality is defined as a property of a system (operator), when its output at time (t) is dependent on its input up to the time (t). The following proposition states integration as an elementary causal operator, [8], [9]:

Proposition 1 *The following are equivalent:*

1. T is causal
2. $P^t T = P^t T P^t$
3. $T = \int dP(t) T P^t$

where P^t is a truncation operator for time (t) and $dP(t)$ is at the time (t). The operation of the integral is written as:

$$T: x \rightarrow y \quad x, y \in H \quad (1)$$

where x is referred to as the domain of T , ($D(T)$) and y is referred to as the image of T , ($I(T)$). H refers to a space where for a given operator T , we have ($\|T\| \leq \infty$).

Definition 1 *The operator T_i is matched with the operator T_j when the $I(T_i) \subset D(T_j)$; or the operator T_j is matched with the operator T_i when $I(T_j) \subset D(T_i)$ [9].*

In models of mechanical elements, force (f) (i.e. force and moments, e.g. a wrench vector) is defined as through variable where its effect is realized through out the elements. Velocity (v) (i.e. angular and linear, e.g. a twist v) is defined as an across variable where a reference frame is needed to determine the relative velocity. These two variables are referred to as fundamental variables. The objective of this section is to model the basic elements of the mechanical system in terms of the fundamental variables as input/output and the notion of causality.

Based on the definition of integral as a causal operator, the following defines models of mass and spring elements:

Proposition 2 *In the causal model of the mass element, force (f) is defined as an input variable and velocity (v) is defined as an output variable.*

Proof: Using the definition of the causal operator, the model of the mass element M is written as a causal operator, or:

$$T_M = \int dP(t) T_M P^t \quad (2)$$

where $T_M \equiv M^{-1}$. Based on the functional relationship of the mass element, its input/output model can be written as:

$$T_M f = \int T_M P^t \int dP(t) = v \quad (3)$$

where for brevity we note that $P^t f = f$. From the above it can be seen that the causal model of the mass element is defined as:

$$T_M: f \rightarrow v$$

where $f, v \in H$.

□

Proposition 3 *In the causal model of the spring element, velocity (v) is defined as an input and force (f) is defined as the output.*

Proof: Using the definition of causal operator, the spring element K is written as a causal operator:

$$T_K = \int dP(t) T_K P^t \quad (4)$$

where $T_K \equiv K$. Based on the functional relationship of the spring element, its input/output model can be written as:

$$T_K v = \int T_K P^t \int dP(t) = f$$

where for brevity we note that $P^t v = v$. From the above it can be seen that the causal model of the spring element is defined as:

$$T_K: v \rightarrow f$$

where $v, f \in H$.

□

Proposition 4 *Damper element can be modelled as having velocity (v) as an input and force (f) as an output or (f) as an input and (v) as an output.*

Proof: In the functional relationship of the damper element, i.e. $f = Cv$, the (f) and (v) are related through a constant C . This representation is not restricted by causal representation and as a result does not restrict the input domain.

□

Proposition 5 *The model of a finger in contact with the grasped object is given as spring and damper elements connected in parallel between the finger and the contact points and the grasped object.*

Proof: The proof is based on the causal models of mechanical elements and the matching condition of the definition (1). Let the model of a finger at the contact point and the model of the object be given as causal operators T_{M_f} and T_{M_o} which representing the two mass elements. When the finger is in contact with the object, we have the following relationship between these causal models of the finger and the object:

$$I(T_{M_f}) \subseteq I(T_{M_o}) \quad (5)$$

From the above it can be seen that the interaction of the two causal operators are not matched, i.e. the interaction is not causal. Let a spring with its corresponding causal model T_{K_c} be attached between finger and the object. The following relationships therefore can be written:

$$\begin{aligned} I(T_{K_c}) &\subset D(T_{M_f}) \\ I(T_{K_c}) &\subset D(T_{M_o}) \end{aligned} \quad (6)$$

As it can be seen, T_{K_c} is matched with both T_{M_f} and T_{M_o} and results in a causal interaction between the fingers and the object.

□

Damper model connected in parallel with the spring model also introduces a causal interaction. The proof is similar to the presence of the spring element.

The compliance can be introduced either through closed-loop control law (i.e. position and velocity gains of the controller) or by attaching compliant material between the finger and the object. For the first method, the closed-loop response of the finger to the deviation from the nominal grasp configuration results in restoring force which is a function of the position and velocity feedback gains of the controller. As a result, the model of the finger is springs and dampers connected between the palm of the hand and the grasped object. In the second method the compliance is introduced in the design of the mechanisms. In this approach, the restoring force is generated through stiffness and damping properties of the compliant material attached for example, between the finger and the object (here we assume that the closed loop compliant property of the finger is negligible comparing to the

properties of the compliant material).

This paper presents two models of the finger based on the second method for introducing compliance. Figure (1) shows the schematics of these two models where compliance is introduced in the design of the mechanisms. In both designs, the contact is modeled as spring and dampers which are obtained from the compliant model of the rubber-type material. In Figure (1a) compliant rubber-type material is attached between the segments of the finger to another solid layer which covers the compliant material. The advantage of this design philosophy is that the interaction between the finger and the object is still causal and in most cases the contact point is modelled as point contact with friction (this type of contact is easier to model and analyze since the contact point does not deform under the applied load). In Figure (1b) the compliant material is attached on the segment of the links without a solid cover.

In the design philosophy of Figure (1a), the solid finger-tip attached through the compliant material to the end-point of the finger creates a spring/damper model of contact. Where, in the design of Figure (1b), the direct attachment of the compliant material to the end-point of the finger creates causal model of contact. However, in the contact model of Figure (1b) in addition to translational forces, the finger can transmit/resist moments about the axis perpendicular to the contacting surface [12], [10].

In both designs it is proposed that force sensors can be located in each segment of the finger near their starting points (Figure (1)). It is proposed that a look up table can be developed such that based on the initial force measurements between the segments and the object, the corresponding contact points can be determined.

For the case of the model of Figure (1b), the look up table also

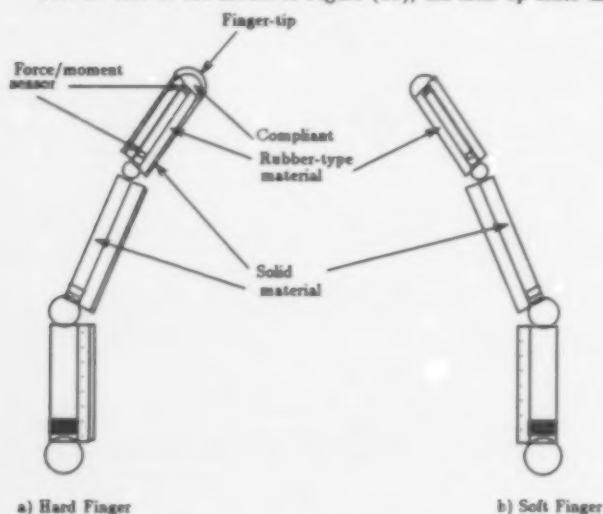


Figure 1: Two models of the compliant fingers

would contain the force/displacement relationships of the compliant material for different types of contact (i.e. vertex, edge and plane contact). These secondary information can be used to further identify the types of contact (Figure 2).

Grasp Stability

This subsection briefly summarizes results of the [11]. It was shown, that if in a force-closure grasp the contacting fingers are modelled to have virtual springs, the resulting grasp is a stable one. That is the grasp object can return to its equilibrium position when it is displaced by an external force (external force is either the unwanted disturbance force or the task force which arises, for example, by plac-

ing an object in a pick-and-place task).

As it was discussed previously, the sources of the virtual springs are assumed to be either through the control servo algorithms or through the presence of soft rubber material between the fingers and the rigid object.

Let the potential energy function of the grasp given the spring models of the grasping fingers (virtual spring) be written as:

$$U = \sum_{i=1}^n \frac{1}{2} k_i x_i^2 \quad (7)$$

where k_i is the equivalent spring property of the i^{th} segment of the fingers in contact with the object in the direction of the displacement

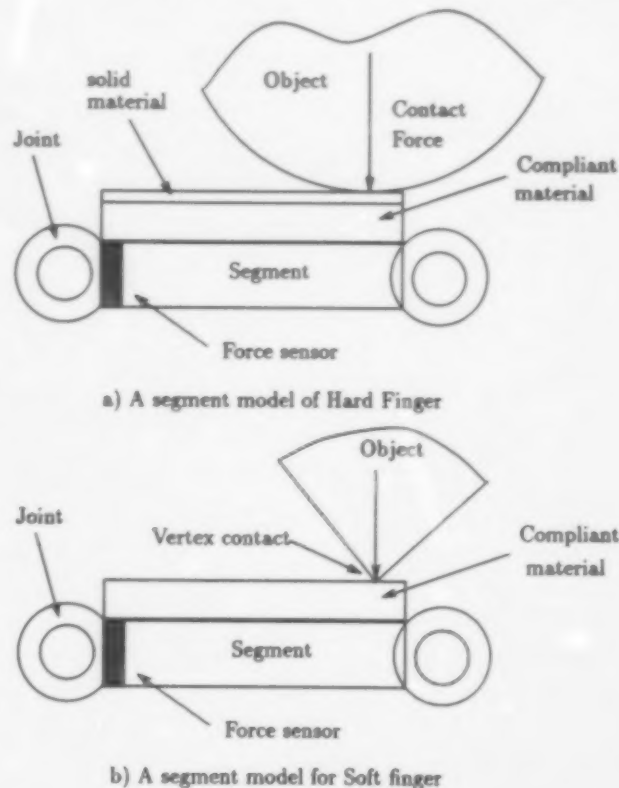


Figure 2: Force/displacement models for the two models of finger x_i , x_i is the displacement of the i^{th} segment from its initial contact configuration with the object (e.g. uncompressed configuration of the spring (e.g. $x_{s=0}$)).

Expanding the function about its equilibrium configuration using Taylor Series expansion we have:

$$U = \sum_{i=0}^n \frac{1}{2} k_i x_{s,i}^2 + x_o^T \nabla U|_{x_{s=0}} + \frac{1}{2} x_o^T \mathcal{H}|_{x_{s=0}} x_o \quad (8)$$

where x_o is a displacement vector (twist vector) of the grasped object expressed in its coordinate frame.

The stability condition of the grasp reduces to the condition of the stiffness matrix K of the grasp with respect to its coordinate frame (which is equal to the Hessian matrix $\mathcal{H}|_{x_{s=0}}$) to be positive definite. In general, the stiffness of grasp object as a function of the spring property of the grasping fingers expressed at the contact frame K_c can be written as: [12] [10]

$$K = GK_c G^T \quad (9)$$

where G is a transformation matrix from the space of grasping forces expressed in the contact coordinate frames to the object reference frame. For a given force closure grasp and a matrix K_c which is diagonal matrix representing the spring constant of the virtual fingers, the relationship of equation (9) represents a positive definite matrix of the grasped object K and hence a stable grasp.

POWER GRASP

This section defined a class of grasp between the compliant fingers and the object where in addition to the finger-tips, some or all the segments of the fingers can make contact with the object [13]. The objective this section is to extend the previous model for the precision type grasp to the modelling of power grasp.

Similar to the previous development [10], for the two models of the finger we can write the following transformation matrix between the forces in a segment coordinate frame and the contact coordinate frame:

$$H = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \text{ point contact with friction} \quad (10)$$

$$H = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \text{ soft contact}$$

Let the force displacement relationship of the grasped object expressed in the contact coordinate frames between the segments and the object be written as:

$$f_s = K_c x_s \quad (11)$$

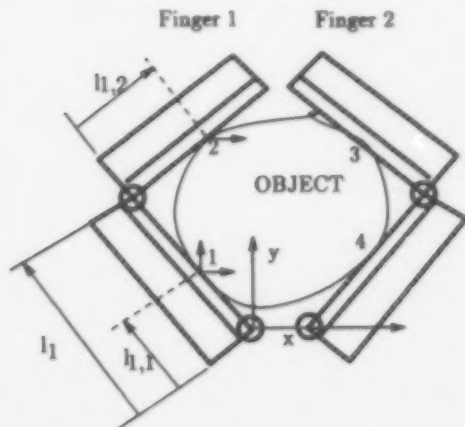


Figure 3: A model of a 2DOF finger in a power grasp configuration

where f_s is a compounded force vector of each segment that is exerting on the object; x_s is compounded vector of the corresponding displacement and K_c is the concatenated stiffness matrix of the compliant material of each contacting segments. For example, in the power grasp model of Figure (3), we are assuming that there exist four contacting points and also object can be displaced in the x and y directions. Therefore, equation (11) can be written as:

$$\begin{bmatrix} f_{s1} \\ f_{s2} \\ f_{s3} \\ f_{s4} \end{bmatrix} = \begin{bmatrix} K_{c1} & 0 & 0 & 0 \\ 0 & K_{c2} & 0 & 0 \\ 0 & 0 & K_{c3} & 0 \\ 0 & 0 & 0 & K_{c4} \end{bmatrix} \begin{bmatrix} x_{s1} \\ x_{s2} \\ x_{s3} \\ x_{s4} \end{bmatrix} \quad (12)$$

where for the example of Figure (3), each of the segment stiffness matrix along the x and y direction of the segment point ($K_{ci} \in \mathbb{R}^{2 \times 2}$).

As it was mentioned before the stiffness of the of each segment at the contact points is assumed to be the stiffness of the compliant material. This stiffness is a function of the grasping forces and the geometry of the contact. For example, Figure (4) shows two schematics for two types of contact and their corresponding force/displacement relationships. As it can be seen, for given grasping force of a finger (internal force), the segment displaces as a function of its spring model. When the grasp is established, the spring model of a segment is identified as it is shown in the Figure (linearized model of the stiffness about grasp configuration).

As it can be seen, for given grasping forces (null space solution of the matrix G [12]), the stiffness of each segment can be determined from its force/displacement relationship given the desired force. This

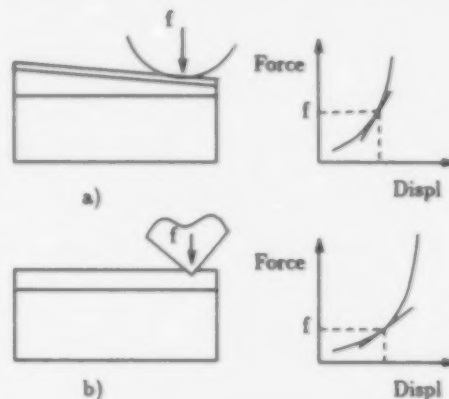


Figure 4: Force/displacement relationship of a segment for two different models of finger

is the feature of the compliant fingers which allows changes in the stiffness of the grasped object as a function of the internal forces (similar discussion has been pointed out by [14]). This feature can be incorporated in the grasp planning algorithm in defining a suitable grasping forces as a function of the desired stiffness of the grasped object.

CONTROL OF GRASPING FORCES

In this section a method for controlling the grasping forces between the segments of the fingers and the object is proposed. It is assumed that for a given grasp configuration, the desired magnitude of the grasping forces are given and the force/displacement relationships at each contact points between the segments and the object are available.

In general, the dynamic model of each finger is a nonlinear function of its joints coordinates. However, during the grasping phase, the linearized model of the fingers about the initial contact points between the segments and the object can be used. Let the linearized model of a finger about configuration parameter which represents the initial contact points between the segments and the object be given as:

$$M_{fi} \ddot{\theta}_{fi} + C_{fi} \dot{\theta}_{fi} + K_{fi} \theta_{fi} = \tau_{fi} + J_{fi}^T f_{ext,i} \quad (13)$$

in the above equation M_{fi} is the mass matrix of the i^{th} finger, C_{fi} is the matrix of the velocity dependent terms, K_{fi} is a matrix of position dependent terms. $f_{ext,i}$ is a vector of the external forces (i.e. grasping forces acting on the segments of the finger).

In general, the grasping forces can be controlled using a centralized controller where the controller uses the measure of all joint positions

and velocities, [15] or by controlling each joint independent of the others. In this paper, a controller based on the later approach is proposed for controlling the grasping forces.

In equation (13), the Jacobian matrix of a finger (e.g. a 2DOF finger) where all its segments are in contact with the object can be written as: (finger #1 of Figure (3))

$$\begin{Bmatrix} \delta x_{s1} \\ \delta y_{s1} \\ \delta x_{s2} \\ \delta y_{s2} \end{Bmatrix}_{f1} = \begin{bmatrix} l_{1,1}\cos(\theta_1) & 0 \\ -l_{1,1}\sin(\theta_1) & 0 \\ l_1\cos(\theta_1) + l_{2,2}\cos(\theta_1, \theta_2) & l_{2,2}\cos(\theta_1 + \theta_2) \\ -l_1\sin(\theta_1) - l_{2,2}\sin(\theta_1 + \theta_2) & -l_{2,2}\sin(\theta_1 + \theta_2) \end{bmatrix}_{f1} \begin{Bmatrix} \delta\theta_1 \\ \delta\theta_2 \end{Bmatrix}_{f1} \quad (14)$$

This is a transformation from the space of infinitesimal displacement of the joints (velocity) to the infinitesimal displacement of the segments at the contact points in the x and y directions. The transpose of the Jacobian which relates the external forces (grasping forces) to the input torques can be written as:

$$\begin{Bmatrix} \tau_1 \\ \tau_2 \end{Bmatrix}_{f1} = \begin{bmatrix} l_{1,1}\cos(\theta_1) & -l_{1,1}\sin(\theta_1) & l_1\cos(\theta_1) + l_{2,2}\cos(\theta_1, \theta_2) & 0 \\ 0 & 0 & l_{2,2}\cos(\theta_1 + \theta_2) & l_{2,2}\cos(\theta_1 + \theta_2) \\ -l_1\sin(\theta_1) - l_{2,2}\sin(\theta_1 + \theta_2) & -l_{2,2}\sin(\theta_1 + \theta_2) & 0 & 0 \end{bmatrix}_{f1} \begin{Bmatrix} \delta f_{x_{s1}} \\ \delta f_{y_{s1}} \\ \delta f_{x_{s2}} \\ \delta f_{y_{s2}} \end{Bmatrix}_{f1} \quad (15)$$

Form the force/displacement relationship of equation (12), the resultant external forces acting on the segments as a function of the joint displacement can be written as:¹

$$\mathbf{J}_{f1}^T \mathbf{f}_{ext1} = \mathbf{J}_{f1}^T \begin{bmatrix} \mathbf{K}_{c1} & 0 \\ 0 & \mathbf{K}_{c2} \end{bmatrix} \mathbf{J}_{f1} \boldsymbol{\theta}_{f1} \quad (16)$$

where for clarity we note $\delta\theta_{f1} = \boldsymbol{\theta}_{f1}$ and also $\mathbf{K}_{c1}$ is the linearized model of the spring property about a given displacement of the segment (see Figure (4)). In this form, the external forces can be combined with the dynamic model of the finger. As a result, by controlling the position of each segment, the corresponding grasping force can be indirectly controlled using the relationship of equation (12).

The position controller proposed here is based on [16], [17] and is written as:

$$\tau_{act f1} = \left[\mathbf{K}_p \quad \mathbf{K}_v \right]_{f1} \begin{Bmatrix} \boldsymbol{\theta} \\ \dot{\boldsymbol{\theta}} \end{Bmatrix}_{f1} + \mathbf{K}_{serv, f1} \boldsymbol{\xi}_{f1} \quad (17)$$

where $\mathbf{K}_{p, f1}$ is the position gain matrix, $\mathbf{K}_{v, f1}$ is the velocity gain matrix, $\mathbf{K}_{serv, f1}$ is the gain matrix on the state of an auxiliary dynamic model representing the external disturbances and $\boldsymbol{\xi}$ is the state of the auxiliary dynamic model of the disturbances define as:

$$\dot{\boldsymbol{\xi}}_{f1} = \mathbf{A}_{f1} \boldsymbol{\xi}_{f1} + \beta_{f1} \mathbf{e}_{f1} \quad (18)$$

where $\mathbf{A}$ is the companion matrix for of a linear differential equation modelling the disturbance and $\mathbf{e}_{f1}$ is a vector of joint position error between the desired joint displacements and the actual ones. The controller of equation (17) has the property that the joint angles of the finger follow the desired values in the presence of disturbances and model uncertainties as long as the closed-loop system remains stable.

¹Here we are assuming that during grasping phase, the object does not move. The condition where the fingers are not simultaneously making contact with the object is a topic of future research.

CONCLUSION AND FUTURE WORK

This paper presented a formal approach for modelling the contact between the fingers of an end-effector and objects. In this model, compliant material is attached between the fingers and object. Two design philosophy were proposed. In the first design, contact between the compliant fingers and object resulted in a point contact with friction where in the second design resulted in a soft contact.

In both designs it was proposed that the location of contact between each segment and the object can be determined through the force sensor measurements located at the beginning of each segments. Look-up tables approach was also proposed for detecting the types of contact (e.g. vertex, edge or line contact for soft finger). It discussed that when the force/displacement relationships between each segment points of the finger are available, the grasping force controller can be reduce to the position control approach.

The future work can be categorized as follow:

- Development of a simple two fingered end-effector based on the two proposed designs for experimenting with the advantages of using look-up tables in identifying the contact types.
- Investigating into the Jacobian relationship of a power grasp in generating the desired grasping forces and the desired stiffness of the grasp.
- Stability analysis of the controller by considering the dynamic model of the whole object/fingers.

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Solving Combined Grouping and Layout Problem Using Simulated Annealing

Attahiru Sule Alfa
Department of Mechanical
and Industrial Engineering
University of Manitoba
Winnipeg, Manitoba

Mingyuan Chen
Industrial Systems Engineering
Faculty of Engineering
University of Regina
Regina, Saskatchewan

Sundresh Heragu
Department of Decision Sciences
and Engineering Systems
Rensselaer Polytechnic Institute
Troy, New York, U.S.A.

In this paper, the simultaneous solution of the machine grouping and layout problem in flexible manufacturing systems is explored. A model for the combined problem is presented. Since the model is complex to solve using traditional optimisation techniques, a suboptimal procedure involving the use of a simulated annealing based algorithm is suggested. Results of numerical examples are presented.

INTRODUCTION

In the design of a flexible manufacturing system, a number of decisions have to be made. Some of the more important ones include (Wemmerlov and Hyer, 1986): type and number of machines required to process a given set (usually, a large number) of parts; type and number of material handling equipment required to transport parts and other material between machines; grouping of machines into their respective cells; layout of machines within cells; and layout of cells with respect to one another.

In order to select the type and number of machines, it is obvious that the type and number of parts to be manufactured in a specific planning horizon, as well as their routings must be known. Part routing refers to the sequence of machines visited by a part. In the redesign of an existing system, the machines to be used are known in the initial phase of the flexible manufacturing system design process. However, while designing new systems, this information will not be available. In the later case, the designer must at least know the sequence of operations required by each part. If this is known and information concerning the machining capability of a set of candidate machines is available, the required type and number of machines may be determined. It has been shown that the layout problem and machine/part grouping problems are NP complete (Vakharia and Wemmerlov (1991)). It is often too difficult to address all the design decisions simultaneously. This is perhaps, the main reason why researchers have addressed these problems independently. In this paper, we will make an effort to address the three decisions: machine grouping; layout of machines within a cell; and layout of cells in a factory, jointly. Thus this paper differs from tradi-

tional papers as it does not assume that the machine grouping and layout problems are independent of one another. The formulation presented in this paper requires that the desired number of groups and the locations of the cells be predetermined.

THE MODEL

The parameters used in the model are defined as follows:

- n : total number of machines in the entire system.
- g : total number of departments in the system ($1 < g < n$).
- y_i : vertical distance of machine i from the point of reference in each department.
- z_i : horizontal distance of machine i from the point of reference in each department.
- c_{ij} : cost per unit distance travelled between machines i and j .
- f_{ij} : frequency of trips between machines i and j .
- l_i : length of the horizontal side of machine i .
- b_i : length of the vertical side of machine i .
- dv_{ij} : minimum distance by which machines i and j are to be separated vertically.
- dh_{ij} : minimum distance by which machines i and j are to be separated horizontally.
- A_p : area of department p , $p = 1, \dots, g$.
- d_{pq} : distance between departments p and q .

$p = 1, \dots, g, q = p + 1, \dots, g.$

C_{pq} : cost per unit distance travelled between departments p and q .

M : a large positive number.

The 0-1 integer decision variables are defined as follows:

$$x_{ip} = \begin{cases} 1 & \text{if machine } i \text{ belongs to department } p; \\ 0 & \text{otherwise,} \end{cases}$$

and let $X = \{x_{ip} | i = 1, \dots, n, p = 1, \dots, g\}$.

In solving this group technology/facility layout (GT/FL) problem, we assume that there are loading/unloading stations in each department and parts will be collected together before they are transported to the other departments.

The GT/FL Model

Let:

$$F_p = \sum_{i=1}^n \sum_{j=1}^n c_{ij} f_{ij} (|y_i - y_j| + |z_i - z_j|) x_{ip} x_{jp} \quad (1)$$

$$F_1 = \sum_{p=1}^g F_p \quad (2)$$

and

$$F_2 = \sum_{p=1}^g \sum_{q=p+1}^g d_{pq} C_{pq} \left[\sum_{i=1}^n \sum_{j=1}^n f_{ij} x_{ip} x_{jq} \right] \quad (3)$$

The 0-1 nonlinear programming model for solving the combined GT/FL problem is presented below.

(GT/FL)

$$\min F = W_1 F_1 + W_2 F_2 \quad (4)$$

s.t.

$$|z_i - z_j| + M u_{ij} \geq \frac{1}{2} (l_i + l_j) + d h_{ij} \quad (5)$$

$$|y_i - y_j| + M (1 - u_{ij}) \geq \frac{1}{2} (b_i + b_j) + d v_{ij} \quad (6)$$

$$\sum_{p=1}^g x_{ip} = 1, \quad i = 1, \dots, n \quad (7)$$

$$\sum_{i=1}^n x_{ip} \geq 1, \quad p = 1, \dots, g \quad (8)$$

$$\sum_{i=1}^n l_i b_i x_{ip} \leq \zeta A_p, \quad p = 1, \dots, g, \quad 0 < \zeta < 1 \quad (9)$$

$$u_{ij} (1 - u_{ij}) = 0 \quad (10)$$

$$x_{ip} \in \{0, 1\} \quad (11)$$

In the above presented model, F_1 is the sum of the cost of parts transportation within each department; F_2 is the cost of parts transportation between departments; W_1 and W_2 are weights associated with the inner and intra department costs, respectively. Constraints (5) and (6) ensure that the machines in each department are separated according to related requirements; constraint (7) ensures that each machine belongs to one group; constraint (8) ensures that each group has at least one machine; and constraint (9) ensures that the total space of a group of machines can be accommodated by the department with parameter ζ to make adjustment. It can be noted that above GT/FL model is an enhanced version of the facility layout model presented in

Heragu and Kusiak (1991). That facility layout model is shown below.

$$(FL) \quad \min F_p \quad (12)$$

s.t.

$$(5) \text{ and } (6) \quad (13)$$

This facility layout model and associated algorithm will be used in this paper to generate optimal or sub-optimal layout within each department p .

The formulation (1)-(11) considers only the case of a simple GT/FL problem with basic constraints. It does not consider special constraints such as: making sure that machines are not located adjacent to each other; keeping some machines in specific locations or making sure that some machines are placed in adjacent locations; or having the minimum number (> 1) or a maximum number ($< g$) of machines in each cell. These can be added by including the constraints in Heragu (1991).

An algorithm based on the simulated annealing approach combined with the heuristic method presented in Heragu and Kusiak (1991) is developed to solve the GT/FL model (Eqs(4) to (11)). The algorithm consists of first using Simulated Annealing (SA) to obtain good preliminary results. Using these results as initial solution to the problem, the modified penalty algorithm presented in Heragu and Kusiak (1991) is then used to solve the problem. A brief description of the Simulated Annealing and the modified penalty method follows.

SIMULATED ANNEALING

Simulated annealing (SA) is a technique based on ideas from statistical mechanics. The main difference between SA and local optimisation is that whereas local optimisation does not accept non-improvements at any stage SA does with a certain probability. Kirkpatrick et al. (1983) proposed this SA as an analogy between statistical mechanics and combinatorial optimisation. They used it to solve a circuit board layout and wiring problem.

SA has also been used successfully to solve traveling salesman problems (Bonomi and Lutten, (1984)), quadratic assignment problems (Wilhelm and Ward (1987)), graph partitioning (Johnson et al. (1989)), plant layout (Heragu and Alfa (1991)) and vehicle routing (Alfa et al. (1991a)).

SA can be described in the following steps (modified from Johnson et al. (1989)):-

Step 1. Get an initial (feasible) solution S randomly.

Step 2. Get an initial temperature (a parameter) $T_m \geq 0$.

Step 3. Do the following N_s times.

3.1 While not yet frozen perform the following.

[The term frozen refers to the state at which costs cannot be improved further].

3.1.1 Find a neighbor S' of S randomly.

3.1.2 Let $\delta = \text{cost}(S') - \text{cost}(S)$.

3.1.3 If $\delta \leq 0$ (downhill move),

Set $S = S'$.

3.1.3 If $\delta > 0$ (uphill move),

Set $S = S'$ with probability $e^{-\delta/T_m}$.

3.2 Set $T_m = R_s T_m$ (reduce temperature).

(Parameter R_s is the cooling rate of SA).

Step 4. Return S .

THE ALGORITHM

SA and Pair-Wise Interchanges

The heuristic algorithm for solving the GT/FL problem is a combination of the simulated annealing procedure and a modified penalty method. The SA is used to generate large number of feasible solutions through pair-wise interchanges of machines in different departments. When the SA procedure ends after a pre-determined large number of total iterations, a modified penalty method is utilised to improve the facilities layout in each of the departments. It can be noted that the SA procedure of the algorithm presented in this section is only one of many possible versions. Discussion of more complicated SA procedures can be found in Alfa et al. (1991a). In addition, alternative feasible solutions are generated by pair-wise interchanges in the algorithm. Other heuristic methods, such as three-way interchanges, can also be used if necessary.

The Modified Penalty Method

The modified penalty method involves transforming the constrained GT/FL model presented in Eqs(4) to (6) into an unconstrained one using the penalty method for each of the departments. This is achieved by first rearranging the constraints in order to have zero's on the right hand side and then squaring the resulting left hand side. These squared terms are multiplied by penalty parameters and then added to the objective function. Detailed discussion of the transformation can be found in Heragu and Kusiak (1991). The unconstrained model obtained is then solved using the Powell algorithm for unconstrained optimisation.

The Heuristic Computational Algorithm

The heuristic algorithm is presented below.

Step 1.

Step 1.0. Set up the values of parameters K, T_m, R_t .

Step 1.1. Separate the machines randomly to generate a initial solution $X^0 = \{x_{ij}^0\}$ which satisfies constraints (7), (8) and (9). Let $X^* = X^0$, $X^a = X^0$, and let $k = 0$.

Step 1.2. For each machine group $p, p = 1, \dots, g$, Based on the initial solution X^0 , calculate total inner department cost $\bar{F}_p = \sum_{i=1}^n \sum_{j=1}^n c_{ij} f_{ij} (|y_i - y_j| + |z_i - z_j|) x_{ij} x_{jp}$ for each department p with a randomly generated feasible layout. Calculate $\bar{F}_1^0 = \sum_{p=1}^g \bar{F}_p$.

Step 1.3. Generate total intra-department cost F_2^0 .

Step 1.4. Generate total transportation cost $F^0 = \bar{F}_1^0 + F_2^0$.

Step 1.5. Let $X^* = X^a = X^0$, $F^* = F^0$, goto Step 2.

Step 2. Let $k = k + 1$, if $k \geq K$, goto Step 9; otherwise, goto Step 3.

Step 3. Generate 2 random integers r_1 and r_2 , $1 \leq r_1 \leq n, 1 \leq r_2 \leq n, r_1 \neq r_2$.

Step 4. If machines r_1 and r_2 are in a same department for solution X^a , goto Step 3; otherwise, goto Step 5.

Step 5. Based on the current solution X^a , exchange machines r_1 and r_2 to generate a new solution X^b . Calculate F^b based on X^b , following the procedure in Steps 1.2 to 1.5.

Step 6. If $F^b < F^*$, let $X^* = X^a = X^b$, goto Step 2. otherwise, goto Step 7.

Step 7. Generate a random number r_3 in $[0.0, 1.0]$. If $\exp[-(F^b - F^*)/T_m] > r_3$, goto Step 2; otherwise, goto Step 8.

Step 8. Let $X^a = X^b$ and let $T_m = R_t T_m$, goto Step 2.

Step 9. With X^* as a starting solution, use the algorithm presented in Heragu and Kusiak (1991) for postprocessor improvement for facility layout for each department $p, p = 1, \dots, g$. Stop.

EXPERIMENTS AND RESULTS

The algorithm presented in Section 4 was tested by a number of numerical examples. Results of two of the examples with 20 and 30 machines grouped into 3, 4, and 5 departments are presented below. Initial feasible solutions for these example problems were generated randomly. Multi-row layout in each of the departments were assumed. Distance matrices of the departments are given in Table 1. Flow matrices of the 20 and 30 machine problems are from Heragu and Kusiak (1991). Relative weights of W_1 and W_2 associated with inner- and inter-department flows are assumed to be 1. Computational results of the two examples are presented in Table 2. In Table 2, F_i , F_m and F_j are the total cost of the initial solution, the cost of the solution from SA procedure (Steps 1 to 8 in the Algorithm) and the cost of the final solution (after Step 9), respectively. For each of the problems, the number of total iterations K was given 10,000. K' is the actual number of the pair-wise interchanges performed since the two machines to be interchanged may be in the same department and such interchanges will not be performed in the SA procedure. It can be noted that in Table 2, simulated annealing can be used to solve the GT problem and the solutions can be further improved by the modified penalty method.

Table 1: Distance Matrices

Dept No.	1	2	3	Dept No.	1	2	3	4
1	0	5	5	1	0	5	5	10
2	5	0	10	2	5	0	10	5
3	5	10	0	3	5	10	0	5
				4	10	5	5	0

Dept No.	1	2	3	4	5
1	0	5	10	5	10
2	5	0	5	10	5
3	10	5	0	15	10
4	5	10	15	0	5
5	10	5	10	5	0

Table 2: Computational Results

No. of Depts.	20 Machines				
	F_i	F_m	F_f	CPU(sec.)	K'
3	2792	2385	2385	19.91	6662
4	3309	2671	2669	26.55	7546
5	3956	3297	3273	30.24	8030

No. of Depts.	30 Machines				
	F_i	F_m	F_f	CPU(sec.)	K'
3	6192	5308	5307	30.93	6628
4	5875	5492	5462	38.87	7529
5	7577	6627	6627	39.50	7974

DISCUSSION

In this paper, an optimisation model for solving combined group technology/facility layout (GT/FL) problem is developed. A simulated annealing based algorithm is proposed to solve the non-linear integer model. The results from the numerical examples are encouraging. This research is an attempt to solve the GT/FL model by using a simple heuristic method. The model and the algorithm could be improved by incorporating various techniques such as:

1. alternative models can be used to formulate the GT/FL problem as proposed in Alfa et al. (1991b);
2. three-way exchange instead of pair-wise exchange of facilities and different values of parameters can be employed in the SA procedure;
3. the modified penalty method can be incorporated into the algorithm for "postprocessing" whenever a new solution is generated by the SA procedure. Better solutions can be expected but the computational time will be much longer.

The latter two alternatives are the topics of future research.

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CONFIGURATION LIFE-CYCLE OF MANUFACTURING CELLS

S. M. Taboun

Department of Mechanical Engineering
University of Ottawa
Ottawa, Ontario, K1N 6N5
Canada

ABSTRACT

In this paper a mathematical model for re-configuring cellular manufacturing systems is developed. The model takes into consideration changes in the production level, termination and allocation of components to the part family, and the cost of handling parts within and between cells. Minimization of material handling costs gives the optimal configuration life-cycle of manufacturing cells. A typical example is illustrated.

1. INTRODUCTION

Cellular manufacturing is a direct application of group technology (GT) philosophy. The basic idea in cellular manufacturing (CM) is to cluster parts that have similar processing requirements into groups known as part families, and machines that meet these requirements into groups known as machine cells. Reduction in material-handling, work-in-process, setup and throughput times, as well as simplified flow of parts and tools, are among the leading benefits of CM which have a profound implications on the overall productivity of a manufacturing organization (Burbidge 1963, 1971, 1975).

Tompkins and Reed (1976) estimate that between 20% and 50% of the total manufacturing expenses can be attributed to material handling. Furthermore, these authors claim that a cost reduction of at least 10% to 30% can be accomplished through effective facilities planning. Effective facilities planning includes achievement of the best machine cell arrangement which can be determined by one of the available heuristic algorithms such as CRAFT and so on, however, the subject of when to re-configure the manufacturing cells is a very important issue. The latter is related to the cell life-cycle and is a function of part(s)/product(s) life-cycle(s). Theoretically, each manufacturing cell is usually dedicated to process a single part family which itself is subject to change

from time to time in terms of part mix and/or production volume. New parts are usually assigned to the part family and production volume of some parts that were initially assigned to the machine cells may decline or discontinue. In many situations, the assignment of new parts and changing part mix of the part family may create some difficulties in confining process requirements to one machine cell. The presence of alternate process plans, duplication of machines and subcontracting may not always eliminate the problem of intercell moves (Rajamani et al, 1990). Thus, the re-configuration problem of machine cells becomes an important issue to be resolved.

A survey of 33 companies conducted by Nicol and Hollier (1983) in Great Britain indicated that the layout stability to be approximately three years and radical layout changes occur frequently in practice. This problem has attracted the attention of several researchers who are focusing their work on the dynamics of facility planning. Recently, Rosenblatt (1986) has focused on job shop re-configuration problems while Afentakis (1986) and Afentakis, et al (1990) have focused on flexible manufacturing systems. Several questions related to the dynamics of facilities planning for manufacturing cells can be addressed such as the following:

- What is the best sequence of cell configurations over a given planning horizon partitioned into time periods?
- When is the optimal point to re-configure manufacturing cells?

The former question has been addressed and studied in relation to the dynamic layout strategies for flexible manufacturing systems (FMS) by Afentakis et al. (1990), while the later question was included in the same study by considering two distinct approaches to determine the point of relayout for FMS. The first strategy was based on a periodic approach where a relayout is performed after Z time units or periods (arbitrary selected) have passed. The second strategy was analogous to an (s,S) inventory policy which is based on similar reasons used in this policy.

The objective of this paper, is therefore, to develop a mathematical model that will determine the optimal life-cycle or re-configuration time of manufacturing cells. Typical production parameters such as the cost of intercell and intracell moves, re-configuration cost, and part life-cycle related variables must be incorporated.

2. ASSUMPTIONS AND NOTATIONS

To simplify model developments several assumptions have been made. These are as follows:

- 1- The life-cycle of each part can be approximated and divided into three periods; the first period represents the start-up and rapid growth phases, the second period represents the maturity and stability phases, and the third period represents the decline phase.
- 2- Production trend of each part is compatible with the above three periods of part life-cycle and continuous in time during each period as shown in Figure 1.

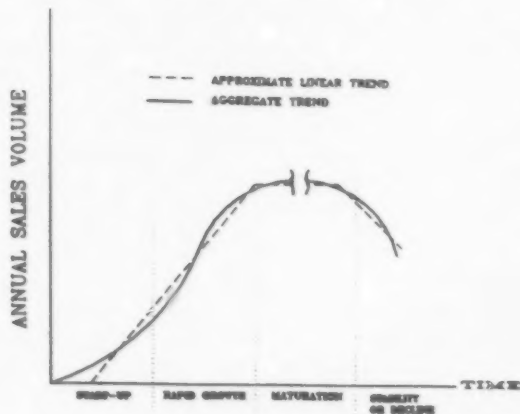


Figure 1. The Product Life-Cycle

- 3- During each production period the rate of change in the production level is constant with respect to time. In other words, trends of the production levels are linear.
- 4- Duration of the above three production periods as well as their corresponding production levels are known and deterministic.

- 5- The manufacturing cell is operated at full capacity and dedicated to process a designed set of parts (part family). Production of the new allocated parts cannot be started-up unless the production rate of one or more components of the part family has started to decline. That is, certain processing capacity is available for use.
- 6- Each cell consisted of a set of machines configured to process a part family. In pure GT cells, components of each part family are not designed to have intercell moves. However, prior to the cell re-configuration, new allocated part(s) may require some intercell moves (i.e. the part is partially processed within the cell) to complete the required operations. After cell re-configuration, similar conditions may occur to the initial set of allocated parts.
- 7- The cost of intercell moves for each part is higher than the cost of intracell moves.
- 8- Duration of the actual cell re-configuration time is very small as compared to the cell life-cycle itself, hence, it can be neglected.

The following notations are used throughout the model developments:

- n Total number of parts processed in the cell prior to its re-configuration.
- N Total number of parts processed in the cell prior to and after its re-configuration.
- Q_j Production level (parts/unit time) of part j during the second production period, where $j = 1, 2, 3, \dots, n, n+1, n+2, \dots, N$.
- α_j Decline rate in the production level of part j during the third production period.
- δ_{n+j} Growth rate in the production level of part $n+j$ during the first production period.
- G_j Duration of the first production period of part j .
- S_j Duration of the second production period of part j .
- D_j Duration of the third production period of part j .
- t Time at which the cell must be re-configured.
- C_{1ij} Cost of intracell moves for part j prior to (i=1) and after (i=2) cell re-configuration.
- C_{2ij} Cost of intercell moves for part j prior to (i=1) and after (i=2) cell re-configuration.
- HCP Handling cost prior to the cell re-configuration within the cycle of interest.
- HCA Handling cost after the cell re-configuration within the cycle of interest.
- CRC Cell re-configuration cost excluding the cost of processing machines and tools.
- TCH Total cost of handling operations without any plans to re-configure

the cell.
TCHR Total cost with plans to re-configure the cell (part handling and cell re-configuration costs are included).

SR Savings in the cost of handling operations due to cell re-configuration.

3. MODEL DEVELOPMENT

Consider a manufacturing cell which is designed to process a part family that has n parts. Production level of part j during the second period is fairly constant at a rate of Q_j (parts/unit time). During the third production period at time T_j , this level of production has started to decline at a constant rate of α_j (parts/unit time) and terminated at time $T_j + D_j$. Meanwhile, a new part $n+j$ is assigned to the cell and production has started-up at time T_j at a constant growing rate of β_{n+j} (parts/unit time). Production of the new assigned part is expected to reach a steady level of Q_{n+j} parts/unit time as shown in Figure (2).

Again, assume that re-configuring the manufacturing cell is justifiable, that is TCH greater than TCHR, and the cost of intracell moves prior to and after cell re-configuration is the same for each part. Hence, it is possible to determine the optimal time at which the cell must be re-configured in order to maximize the savings in handling costs.

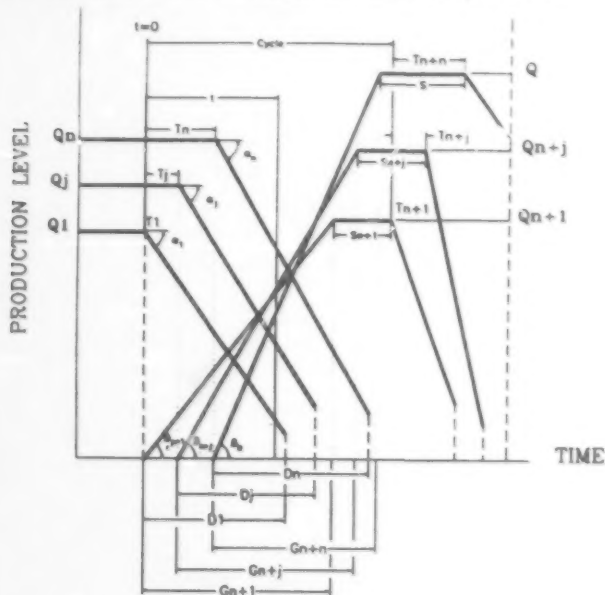


Figure 2. Production Trends of Manufacturing Parts Before and After Cell Re-configuration

The total cost of handling operations prior to cell re-configuration is as follows:

$$HCP = \sum_{j=1}^n C_{11j} [Q_j T_j + Q_j (t - T_j) - \alpha_j (t - T_j)^2 / 2] +$$

$$\sum_{j=1}^{N-n} (C_{11,n+j} + C_{21,n+j}) \beta_{n+j} (t - T_j)^2 / 2 \quad (1)$$

The cost of handling parts after re-configuring the manufacturing cell is as follows:

$$HCA = \sum_{j=1}^n \{ (C_{12j} + C_{22j}) [(D_j + T_j - t) (Q_j - \alpha_j D_j) + \alpha_j (D_j + T_j - t)^2 / 2] \} + \sum_{j=1}^{N-n} C_{12,n+j} \beta_{n+j} \times [(G_{n+j} - T_j)^2 / 2 - (t - T_j)^2 / 2] + (S_{n+j} - T_{n+j}) Q_{n+j} \quad (2)$$

The total cost of handling operations and cell re-configuration is as follows:

$$TCHR = HCP + CRC + HCA \quad (3)$$

Differentiating equation (3) with respect to time, t and equating it to zero gives the following:

$$\begin{aligned} \frac{d}{dt} (TCHR) &= \frac{d}{dt} (HCP) + \frac{d}{dt} (CRC) + \frac{d}{dt} (HCA) \\ &= \sum_{j=1}^n C_{11j} [Q_j - \alpha_j (t - T_j)] + \sum_{j=1}^{N-n} (C_{11,n+j} + C_{21,n+j}) \beta_{n+j} (t - T_j) + \sum_{j=1}^n (C_{12j} + C_{22j}) [(Q_j D_j - Q_j) - \alpha_j (D_j + T_j - t)] - \sum_{j=1}^{N-n} C_{12,n+j} \beta_{n+j} (t - T_j) = 0 \end{aligned} \quad (4)$$

Equation (4) can be rearranged as follows:

$$\begin{aligned} \sum_{j=1}^{N-n} [C_{21,n+j} + (C_{11,n+j} - C_{12,n+j})] \beta_{n+j} (t - T_j) &= \sum_{j=1}^n [C_{22j} + (C_{12j} - C_{11j})] \times [Q_j - \alpha_j (t - T_j)] \end{aligned} \quad (5)$$

Hence, the optimal re-configuration point can be determined as follows:

$$t = \left\{ \sum_{j=1}^n [C_{22j} + (C_{12j} - C_{11j})] (Q_j + \alpha_j T_j) + \sum_{j=1}^{N-n} [C_{21,n+j} + (C_{11,n+j} - C_{12,n+j})] \beta_{n+j} T_j \right\} /$$

$$\sum_{j=1}^n [C_{22j} + (C_{12j} - C_{11j})] \alpha_j + \sum_{j=1}^{N-n} [C_{21,n+j} + (C_{11,n+j} - C_{12,n+j})] \beta_{n+j}$$

$$N-n \sum_{j=1} [C_{21,nj} + (C_{11,nj} - C_{12,nj})] B_{nj} \quad (6)$$

Again, if the difference of the intracell moves cost prior to and after cell re-configuration for each part were very small and close to zero, then equation (6) becomes as follows:

$$t = \frac{\sum_{j=1}^n C_{22j} (Q_j + a_j T_j) + \sum_{j=1}^{N-n} C_{21,nj} B_{nj} T_j}{\sum_{j=1}^n C_{22j} a_j + \sum_{j=1}^{N-n} C_{21,nj} B_{nj}} \quad (7)$$

Equations (6) and (7) give the optimal point of time at which the manufacturing cell must be re-configured in order to maximize the savings of the handling operations cost. Substituting for $n=1$, $N=2$ and $T_1=0$ in the above equation leads to the optimal time of a single part case.

4. NUMERICAL ILLUSTRATION

A manufacturing cell is designed to process three parts A, B, and C at production rates of 8,000, 10,000 and 13,000 parts/month respectively. These rates are expected to decline at $T_A=0$, $T_B=6$ and $T_C=14$ months with constant rates of 275, 425 and 780 parts/month respectively. New parts D, E and F are expected to be allocated to the cell as soon as the production of parts A, B and C begin to decline to improve cell utilization. Production of the new assigned parts (A, B, and C) is expected to increase at constant rates of 425, 590, and 365 parts/month respectively for at least 26 months for part A and 22 months for parts B and C, then becomes fairly stable. Parts A, B and C are expected to discontinue when production level of each one reaches a level of 1,000 parts/month. At which point the manufacturing cell should be re-configured to accommodate the production of the new assigned parts and minimize handling costs? The relevant costs of both intracell and intercell moves for each part are as follows:

$C_{1A} = \$1.0$	$C_{2A} = \$1.5$	$C_{1B} = \$1.1$
$C_{2B} = \$1.7$		
$C_{1B} = 1.2$	$C_{2B} = 1.8$	$C_{1E} = 1.3$
$C_{2E} = 2.0$		
$C_{1C} = 1.5$	$C_{2C} = 2.2$	$C_{1F} = 1.4$
$C_{2F} = 2.1$		

Solution:

Assume that the cost of intracell moves prior to and after cell re-configuration for each part remains fairly the same. Using the above data and substituting in equation (7) we obtain the following:

$$t = \frac{12,000 + 22,590 + 52,624 + 7,080 + 10,731}{2,669 + 2,893.5} = 18.88 \text{ months}$$

Therefore, the cell must be re-configured after 18.88 months from the time that production of part A has started to decline or after 4.88 months from the time that production of part C has started to decline.

5. CONCLUSIONS

In this paper a mathematical model of manufacturing cell configuration life-cycle was developed. The main objective of this model is to determine the exact re-configuration point for the manufacturing cell and the optimal cell configuration life-cycle. The model takes into consideration components of the part family that are declining or no longer in production, while several new components are assigned to the part family to be processed within the system. Configuration life-cycle model of manufacturing cells is based on the life-cycle of each part or product and each is divided into three production periods and approximated using piece-wise linear approximation. Production parameters for each period and handling costs of intercell and intracell movement of parts were incorporated in each model. An example problems is illustrated and solved using the developed model. Other costs such as machining costs can be easily incorporated in the developed model and the optimal cell life-cycle can be determined.

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ADJUSTABLE SPEED DRIVE OWNER EXPERIENCE

David Grier, P.Eng., Director
Tony Kaminski, P.Eng., Ph.D., Senior Design Engineer
Pauline Patenaude, B.Comm, Marketing Specialist

Equipment and Machinery Development Division
Saskatchewan Research Council
15 Innovation Blvd., Saskatoon, Saskatchewan S7N 2X8
Telephone: (306) 933-8131; Fax: (306) 933-7896

ABSTRACT: This paper provides user experience information that was collected via a survey of Adjustable Speed Drive users. It reports that this technology is well received in many applications, but suffers from a bad reputation and lack of information about how to size, operate and maintain the equipment.

1.0 INTRODUCTION

The Canadian Electrical Association (CEA) has identified the potential contribution that adjustable speed drives (ASDs) could make to demand side management programs. However, some market resistance to incorporating ASDs into industrial and commercial applications has been noticed. The CEA commissioned the Saskatchewan Research Council to conduct a survey of ASD users that would:

- establish the type and frequency of problems associated with ASDs;
- establish the cost of resolving these problems;
- establish the problems and causes as a function of region, level of engineering design, installing contractor, type of filtering, age and system type;
- analyze the findings, including performing some statistical analysis on the data and develop general characterizations of the maintenance and repair costs inherent in the use of ASDs.

2.0 METHODOLOGY

The project methodology included the identification of users, the selection of issues to be addressed, the analysis of the survey data and the development of conclusions and recommendations from the analysis. Manufacturers, distributors and dealers of ASDs and utility personnel were contacted for names of known ASD users. Additional users were identified by contacting over 1000 companies from industry that were considered to have applications that could use ASDs. 327 ASD users and were willing to participate in the survey were identified.

A brief review of technical papers and industry articles was conducted to identify the issues that appeared to be of concern. Then, a number of users, manufacturers and suppliers were interviewed by telephone to expand on the information base provided by the literature. Finally two focus group meetings were held in Vancouver to present the issues uncovered to that point, an initial draft of the survey and an overview of the methodology. Participants in each session (industrial users in one and commercial users in the other) provided feedback. As a result, some issues were added and others dropped and some changes were made to the questionnaire format. This three step process greatly increased the potential that the survey would be addressing the right issues in a way that the surveyed companies would respond with meaningful information.

A combination telephone/mail survey methodology was designed that solicited the company's overall impression of ASD technology and requested specific information on individual ASDs used by the respondent. The questionnaire was tested in face to face meetings with six Saskatchewan companies and the entire mail/phone methodology was tested with eight British Columbia companies. The questionnaire was mailed to the 327 users. Quebec companies (100) received questionnaires in both official languages. The contractor followed up with telephone reminders and offers of assistance. A total of 129 companies responded to the survey (40% return rate) reporting on 607 drives.

3.0 DISCUSSION OF FINDINGS

ASDs were well accepted by the companies surveyed with 96.7% stating that an ASD was the most appropriate drive for their application. The harmonic problems

repeatedly reported in the literature were not considered to be a problem by ASD users. Interest in ASDs is very strong in those companies that had not yet tried ASDs and even in those that had removed them because of technical problems.

Users were much more difficult to identify in eastern Canada. More than 60 eastern companies contacted had previously owned ASDs and had removed them, while others had not tried them because they had heard about problems. It appears that improved technology was introduced later in the west, leading to better market acceptance. This theory could not be supported or refuted by the survey because only those that currently had ASDs were surveyed. Statistically, there was no difference between the experience of present eastern and western ASD users.

Drive Reliability, Downtime and Repair Cost

The data showed that 38.5% of ASD users surveyed experienced no problems with their drives. Standard induction motors tend to be reliable pieces of machinery and reliability studies show motor failure rates typically less than 0.1 units per year. Since each ASD has an inverter which was less reliable than the motor it is obvious that an ASD will be less reliable than a standard induction motor. Mean Time Between Failures (MTBF) for motors was calculated to be 6.46 years and was 2.64 years for inverters.

The data showed that 52.1% of the induction motors used in ASDs had no failures. Of the motors which failed, the majority, 54.3%, of the motors operated over two years between failures. This data shows that only 34.1% of all the inverters used on ASDs had no failures. Of the inverters which failed, the majority, 56.6%, failed within one year and only 25% of the inverters operated over two years without failure. Hence it can be concluded that the average time between motor failure is at least double the average time between inverter failure, supporting the MTBFs calculated above.

The frequency of motor and inverter failures was higher in the pulp and paper industry. However, pulp and paper users indicated shorter downtimes than other ASD users.

Users in commercial applications experienced fewer problems with ASDs than those in industrial applications. ASDs in commercial applications were newer than those used in industrial applications. The average age of the motors was older than the average age of the inverters. Drives located in western Canada, especially British Columbia, experienced somewhat fewer problems than drives in eastern Canada. Drives in western Canada were slightly newer than drives in eastern Canada.

In the majority of inverter failures (58.0%) and motor failures (64.1%) the downtime was less than one day. The average downtimes for repair of inverters and motors were similar at 3.2 and 3.1 days respectively.

The repair costs for the majority of motor failures (69.0%) and inverter failures (54.9%) was below \$1,000. While it can be concluded that repair costs of motors and inverters were generally similar, it should be pointed out that repair costs above \$10,000 were more commonly associated with motors than inverters. When all drives are considered (those that failed and those that did not) the average repair cost for motors is \$1,245 and for inverters is \$1,197. It must also be remembered that inverter failures occurred more than twice as often as motor failures. The repair costs per year reported in Table 3 are the average repair costs divided by the calculated MTBFs.

The data in Table 1 shows that the motors were generally older than the inverters. The statistical analysis revealed that there was no relationship between the age of the equipment and its reliability or maintenance requirements.

TABLE 1 ASD RELIABILITY AND MAINTENANCE INFORMATION

	Motor	Inverter
Total Units	607	607
Number That Never Failed	316	207
Average Age of Units Which Experienced Failures	9.2 years	5.1 years
Estimated Average Time Between Failure for all Units	6.46 years	3.65 years
Estimated Frequency of Failures (unit-years)	.155	.378
Average Downtime for Repair	3.1 days	3.2 days
Average Cost to Repair Failed Unit	\$2,620	\$1,940
Average Repair Cost for all Drives	\$1,245	\$1,197
Average Repair Cost Per year	\$193	\$452

TABLE 2 CAUSES OF DRIVE FAILURES

Cause of Failure	ASD Users (Present Study)		Standard Induction Motor Users (EPRI Study)	
	% That Rated Cause Serious	Rank	% of Utility Users	Rank
Inadequate Instruction	16.3	1	-	-
Poor Design	14.7	2	39.0	1
Misapplication	14.0	3	12.9	4
Insufficient Start-up Torque	12.4	4	-	-
Mis-operation	10.2	5	21.5	3
Poor Workmanship	10.1	6	26.6	2
Harmonics*	9.3	7	-	-
Lack of Technical Support*	5.4	8	-	-
Component Selection	3.9	9	-	-
Inverter Sensitivity*	3.9	9	-	-
* Denotes causes of failure which were not listed on the ASD questionnaire but were written in by respondents.				

User Rating of ASD Problems

The problems considered most serious (rated **Very Serious** and **Serious**) by drive users listed in decreasing order of importance were: inverter or motor tripping (31.0% of users), failure of control instrumentation (14.7%), and component failure (8.5%). Harmonic problems such as flickering of lights, high pitched noise, humming of telephone or radio frequency interference was considered a serious problem by about 1% of the users.

The most serious causes (rated **Very Significant** and **Significant**) of ASD problems as rated by users are provided in Table 5. Also shown in the same table are the causes of standard induction electric motor failures which were identified in an Electrical Power Research Institute (EPRI) study of utility users. In the EPRI data shown in Table 2, only four causes of motor failures were provided to users who had to select one cause which contributed most to their motor failures. Hence the addition of the percentages shown was equal to 100.0. In the ASD questionnaire, eight specific causes of ASD problems were listed with space provided for write-in causes. The user was asked to rate the causes very significant, significant, slightly significant or not significant at all. Serious causes of problems were those which users rated very significant and significant. A number of write-in causes of ASD failures were provided by respondents. The percentages of users who rated the write-in causes would probably have been higher if these causes had been listed on the ASD questionnaire for all to consider.

A review of the data in Table 2 on standard induction motors shows that causes of failure listed in order of importance were:

poor design, poor workmanship, mis-operation and misapplication. These four causes of induction motor failure were also felt by users to be major causes of ASD failures with misapplication being ranked higher by ASD users than standard induction motor users.

A comparison of the EPRI induction motor failure data with data obtained in this study on variable frequency inverter drives indicates that with ASDs two new major causes of problems were identified by users: inadequate instruction and insufficient start-up torque. It can be concluded that suppliers of ASDs have not provided users with sufficient information on their drives. Also, many users did not realize that with ASDs there can be considerably less motor start-up torque, especially at lower motor speeds, which causes motor problems such as tripping and various inverter problems. Other causes of ASD motor problems provided by users were: harmonics, lack of technical support, and inverter sensitivity. Even though harmonic effects on lights, telephones or radios were not considered serious, the users did perceive that harmonics associated with the power supplied by the inverter can cause motor problems such as overheating, tripping, and component failures. Two items, lack of technical support and component selection, could be classified with inadequate instruction, making it even more clearly the most important cause of ASD problems.

Analysis of ASD Problems

The technical concerns relating to the widespread use of ASDs that were put forward in the Request for Proposal (RFP) that led to this study were:

- harmonics and their effect on power quality
- drive reliability
- radio-frequency interference
- telephone interference
- acoustic noise and vibration
- motor overheating and excessive voltage stress on windings

All but one of the above concerns, drive reliability, were associated with harmonic effects produced by the inverter. Data obtained from user questionnaires indicated that harmonic effects were of minor concern and produced very few problems. A review of the problems associated with standard induction motors is shown in Table 3. Data in Table 3 shows that bearings and stators are the most common failures in standard induction motors.

In comparing the data in Table 3 on failure of motors in ASDs with failure of standard induction motors it is concluded that the addition of an inverter to vary the frequency of AC power and produce an adjustable speed induction motor drive creates two major new motor problems: tripping and overheating. Tripping was the most common ASD motor problem exceeding both bearings and stators.

Cross tabulations and log linear analysis were done on questionnaire data to determine what parameters affected ASD problems. If a parameter increased the frequency of an ASD problem by less than 50% then this parameter was assumed to have "slightly increased the problem". If a parameter increased the frequency of a problem by more than 50% then the parameter was assumed to have "significantly increased problem". It was noted in a few situations that statistically no conclusions could be made because of insufficient data. However some observations could be made in these situations and they were noted.

Based on statistical analysis, some of the parameters which significantly increased motor failures included: large motor size, high speed of rotation, outside locations, use of arbitrary oversizing guidelines and when no instructions were provided. Other parameters which slightly increased motor

problems included: high torque motor designs, high efficiency motors, continuous operation, more frequent speed variation, direction reversals, auto restart on motor, use of starting torque-boost inverter, having the ASD purchased from a manufacturer or as part of a package, voltage variations in incoming power and application in pulp, paper and petroleum industries. Some of the parameters which significantly increased inverter failures included: increasing motor size, outside locations, intermittent service, direction reversals and arbitrary oversizing guidelines. Other parameters which slightly increased inverter problems included: high torque motors, increasing motor speed, ASD purchased from a manufacturer or as part of a package, high start speed, auto restart on motor, use of microprocessor based on starting torque-boost inverters, no voltage protection on motor and applications in pulp, paper and petroleum industries. Some of the parameters which significantly increased both motor and inverter failures were large motor sizes, outside locations and arbitrary oversizing guidelines.

ASD problem relationship data showed that parameters which increased both motor tripping and overheating problems include: decreasing motor speed, decreasing motor size, high torque motors with service factors above 1.15, operation in non-heated inside locations, constant torque applications, no instructions provided, arbitrary oversizing of drive, and use of microprocessor-based inverters. Parameters that increased both motor bearing and stator problems included: larger motor size, higher motor speed, outside location, auto restart on motor, high start speed, and having an ASD purchased as part of a package.

ASD users experienced significantly more problems with inverters than with their motors. Parameters which increased the major inverter problems, namely, component failures, tripping, blown fuses and adjustment included: increasing motor size, high motor speed, outside locations, auto restart on motor, use of torque-boost inverters and high motor start speeds. Inverter component failures also increased

TABLE 3 FREQUENCY OF ASD MOTOR PROBLEM

Motor Problem	% of ASD Users (Current Study)	% of Induction Motor Users (EPRI Study)
Tripping	20.6	-
Bearings	18.6	41.0
Stator	10.9	37.0
Overheating	10.9	-
Vibration	1.0	-
Rotor	0.8	9.0
Other Motor Components	-	14.0

with high torque motor and in application requiring reversals of motor direction.

During personnel interviews with some ASD users, indications were given that newer drives and drives from offshore sources such as Japan were more reliable. However, the users who filled out questionnaires had such a large number of makes of motors and inverters which were purchased over many years, that it became difficult to statistically show any significant relationship between those parameters. There was some indication that newer ASDs were more reliable and that some makes were somewhat better than others.

Information acquired in the initial phases of this study indicated there were a number of parameters which were thought to effect ASD problems. However, data from the analysis showed that these parameters did not significantly affect ASD problems. The parameters included:

- motor or inverter manufacturers
- drive application, such as constant horsepower drives which have high torque requirements
- selection of the drive
- percentage of total power used by ASD's in a facility

4.0 SUMMARY OF CONCLUSIONS AND RECOMMENDATIONS

ASDs were well accepted by the companies surveyed, with 96.7% stating that an ASD was the most appropriate drive for their application. However, there are some problems being experienced by ASD users with tripping of motor or inverter being the most common.

Inadequate instruction was by far the most common cause of ASD problems. There is an element of inadequate instruction in five of the other causes including misapplication, insufficient start-up torque, mis-operation, improper installation and lack of technical support. Inadequate instruction and lack of technical support was especially important in commercial installations which generally had less qualified technical personnel. ASDs are significantly more complicated than standard induction motor drives. ASDs located in facilities with computers or considerable instrumentation had fewer problems indicating that familiarity with microprocessor based equipment was an asset to successful operation of ASDs.

Users experienced more ASD problems with larger, higher speed motors which were located outside. The motor starting torque decreases with decreased motor speed for a given motor. For a given motor start speed of, say 500 rpm, a 3600 rpm motor will have much lower starting torque than a similar

1800 rpm motor. ASD sizing can be more critical for high speed motors and also for higher horsepower motors. Harmonics present in the power supplied to the ASD motor will decrease motor efficiency and increase overheating problems. Motor overheating increased with lower speed motors which had higher service factors.

Inverter and motor sizing should be based on the required motor torque. A number of the ASDs which were surveyed had motors which were considerably older than the inverters. This would indicate that a constant speed drive was converted to a variable speed drive by adding an inverter. The drive modification would reduce both the motor start torque and also the motor efficiency. If a completely new drive was installed, the motor would be sized based on starting torque required and this could result in a significantly larger motor and inverter for the application which would minimize ASD problems.

Modern inverters with microprocessors offer numerous features and are much more complicated than a standard induction motor control. ASDs are usually integrated with control systems making the whole system too complicated for the average electrical maintenance worker. Lack of system understanding and improper programming can result in tripping problems and component failures. ASD suppliers and users need to have adequate instruction and technical support for the equipment.

It was concluded that present ASD technology performs the function of adjusting the speed of electric motor drives well. It suffers from a bad reputation from older technology and lack of understanding of how to implement, operate and maintain present technology. Increased level of promotion of the advantages of ASDs and the successful experience of many ASD users would deal with the reputation problem. Training activities are key to the continued success of present and new ASD users. Utilities, manufacturers and dealers should all provide increased training opportunities in the following areas:

- application design and ASD sizing
- programming and operation
- problem diagnostics

Manufacturers need to improve the user friendliness of the programmable control systems and to develop model identification methods that refer to key sizing characteristics, such as starting torque, to reduce the temptation to size based on horsepower. A program that permitted a company to trade in an old motor allowing the use of a properly sized motor in a retrofit ASD application would also reduce problems.

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TECHNICAL SESSION 4

AGRICULTURE

Chair: R.L. Kushwaha



LOCAL AIR SPEED WITHIN BULK GRAIN UNDER FORCED VENTILATION

Deqiang Gu
Grad. Res. Assoc.

S. Sokhansanj
Professor

Department of Agricultural Engineering
University of Saskatchewan
Saskatoon, Saskatchewan, Canada S7N 0W0

ABSTRACT

The magnitude of the interstitial air velocity in bulk grain is usually calculated by dividing the superficial air velocity by the bulk porosity of the grain. In the experiment described in this paper, the air velocity measured by hot-wire anemometers embedded in a bed of wheat were consistently higher than the calculated velocity. Additional measurements of the local air velocities using a bare sensor in wheat and field peas confirmed the earlier observation. It was concluded that the magnitude of actual local air velocity within a grain bed may be two to three times the magnitude of the calculated air velocity.

INTRODUCTION

In grain drying literature, the air velocity, V , within a grain bed is calculated by:

$$V = Q/(AE) = V_0/\epsilon \quad (1)$$

where Q is the volumetric flow rate (m^3s^{-1}), A is the frontal area (m^2), and V_0 is the superficial or approaching air velocity ($\text{m}^3\text{s}^{-1}\text{m}^2$). The bulk porosity, ϵ , is the volumetric void ratio of the bulk grain. A common way to determine the bulk porosity is to measure the bulk density, ρ_b , and the particle density, ρ_s , of the grain and use these values in Eq. 2:

$$\epsilon = 1 - \rho_b/\rho_s \quad (2)$$

ASAE Standard D241.2 lists values of bulk porosity for wheat from 0.396 to 0.426 for three wheat varieties. According to Eq. 1, the air velocity, V , within a wheat bed is approximately 2.35 to 2.52 times V_0 .

In an experiment to investigate the airflow distribution in a deep bed of wheat, the hot-wire anemometers embedded in the bulk consistently showed a higher air velocity than that calculated using Eq. 1. A number of tests were then performed to confirm the observation. This paper describes the experiments and discusses the resulting air velocity data.

BACKGROUND

The division of V_0 by ϵ in Eq. 1 stems from the "Dupuit relation" (Carman 1956) or

the "Dupuit assumption" (Dullien 1975). The Dupuit relation or Dupuit assumption states that if the pore spaces in a bed of particles are isotropically and randomly distributed, then the porosity of any layer of infinitesimal thickness normal to the direction of flow would be equal to the bulk porosity, ϵ , of the bed. Chen and Hsueh (1987) called this porosity the "surface porosity".

The ratio V_0/ϵ was called the "average" pore velocity by Dullien (1975). He pointed out though that this value is not equal to the average pore velocity obtained by correctly averaging the velocity over a number of interconnected, differently-shaped and sized pores. Therefore the "Dupuit relation" or "Dupuit assumption" is physically correct only for the special case of uniform shape and size pores.

Carman (1956) stated that the path of an element of fluid is tortuous, and hence the true magnitude of the pore velocity must be higher than V_0/ϵ . He described the flow models for beds of granular particles based on conduit flow, and related the pore velocity V_p to Darcy's superficial velocity V_0 as:

$$V_p = V_0 L_e/\epsilon L \quad (3)$$

where L_e is the actual path length of flow, and L is the length of the bed. The ratio L_e/L is the tortuosity. Carman listed values of L_e/L ranging from 1.41 to 1.71 for beds of various size and shape particles. The bulk porosity of these particles ranged from 0.368 to 0.470.

In deriving the relationship between the pressure drop and the fluid velocity in packed beds, Ranz (1952) analyzed the bottom layer of a model bed composed of spherical particles of uniform size to determine the magnitude of the pore velocity V . These particles were packed with their centres at the corners of tetrahedrons, which gives the closest packing with uniform spheres. He derived an approximate relationship in the form of

$$V/V_0 = 10.73 \quad (4)$$

to relate the superficial velocity, V_0 , to the actual pore velocity, V , of fluid in the bed.

Garrett and Brooker (1965) applied the theoretical analysis of Ranz (1952) to predict the pressure drop for a number of agricultural grains. Based on the measured drag coefficients, they calculated the air velocity using Eq. 1, and predicted the pressure drops that were several times smaller than that measured by Shedd (1953) for the same superficial air velocities. Garrett and Brooker suggested the following ratios relating the superficial air velocity, V_0 , to the air velocity, V , within the following bulk grains:

$V/V_0 = 9.1$	for corn,
$V/V_0 = 7.8$	for oats,
$V/V_0 = 4.2$	for wheat.

To the knowledge of the authors, no research other than Garret and Brooker has addressed the issue of air velocity within the grain mass. Perhaps difficulties in measuring the air velocity within grain bulk has hindered the efforts.

EQUIPMENT AND INSTRUMENTATION FOR WHEAT BED MEASUREMENTS

Fig. 1 is a schematic diagram of the experimental equipment used to measure the air velocity in bulk wheat. The equipment consisted of a laboratory bin with rectangular cross section and the instrumentation for measuring air velocity and static pressure. The bin, made of plywood, was 2.0 m long, 1.0 m wide and 1.5 m high. The floor of the bin is covered with perforated planks, and the open area of the floor is about 12% of the total floor area. A plenum under the floor, 0.4 m deep, extends the length and width of the bin. A variable speed centrifugal fan is connected to the plenum via a 2-metre long circular duct and a transition. A Pitot tube grid is inserted in the duct to measure the air flow rate. The Pitot tube grid was calibrated with a bench-top wind tunnel (TSI Model 8390, TSI Inc., St. Paul, MN.) before installation.

The local air velocity in the bulk wheat was measured by omnidirectional hot-wire

1. Variable speed blower
2. Air duct
3. Pitot tube grid
4. grain bin
5. Air velocity transducer
6. Pressure tube
7. Plenum
8. Power supply & module for anemometer
9. Computer
10. Pressure transducer & indicator

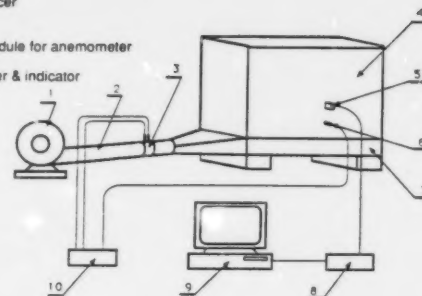


Fig. 1 Schematic diagram of experimental equipment with rectangular bin

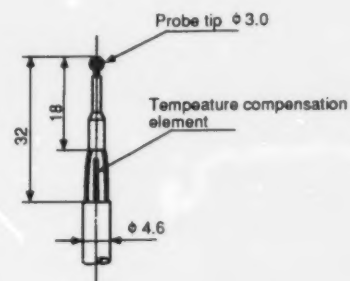


Fig. 2 Details of spherical sensor (TSI Model 8470)

anemometers (TSI model 8470, Fig. 2). The measurement system included the anemometers, a power supply/interconnect module (TSI model 1623), a personal computer and an A/D interface (TSI model 880262, 880263). The air velocity measurement range of the anemometers is 0.0-0.5 m/s or 0.0-1.0 m/s. There is a temperature compensation element on the sensor as shown in Fig. 2. The specified accuracy of the transducer is $\pm 10\%$ of the reading over a 270° solid angle. Since the tip of the hot-wire anemometer is fragile, a protective cover was constructed. The cover is made of stainless steel tube with outside diameter of 8.0 mm, and 85 mm long. On the cover a number of long slots with dimensions of 1.5 mm x 10 mm are machined. The open area of the cover is about 65% of its total area.

Each individual anemometer was calibrated in the wind tunnel with and without the protective cover. The results of calculation indicated the linear relationship between the air velocity in the wind tunnel and the output of the anemometers. The calibration slope was 1.0 for the sensors without the cover. For the covered sensors, the calibration slope varied from 0.7335 to 0.8811.

Rigid stainless steel tubes were inserted

into the grain bed and used to measure the static air pressure. The outside diameter of each tube is 3.2 mm and the inside diameter is 1.3 mm. The tubes were connected to a multi-port valve (scanivalve Model W0601/1P-12T) with 3.2 mm i.d. clear plastic tubes. The multi-port valve provided a means of multiplexing pressure tubes to a single differential pressure transducer (Model DP103-26, Validyne Engineering Corp. Northridge, CA). The measurement range of the pressure transducer is 0-3488 Pa. The transducer was calibrated against a micro-manometer (model 34FB2, The Meriam Instrument Corp. Cleveland, Ohio) which had an accuracy of ± 0.246 Pa. The resolution of the voltage output of the transducer was equivalent to 0.35 Pa.

Fig. 3 shows the locations and the distances between the sampling points for air velocity and static pressure measurements on the wall of the bin. There were nineteen pressure sampling points and ten air velocity sampling points on the wall. The nineteen pressure tubes and the ten anemometers were inserted into the grain mass and had their tips in a vertical plane about 280 mm away from the bin wall. One pressure tube was inserted in the plenum to measure the static pressure of air in the plenum.

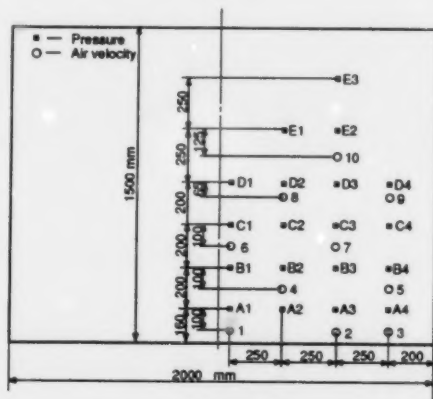


Fig. 3 Location of sampling points for air velocity and pressure transducers on the wall of experimental bin

The experimental bin was filled with freshly harvested wheat of the Katepwa variety. The moisture content of wheat was 14% w.b., measured by the standard oven method (ASAE S352.2). The bulk density of the wheat ($\rho_b = 802 \text{ kg/m}^3$) was measured with a 0.5 liter standard container. The kernel density of the wheat ($\rho_s = 1419 \text{ kg/m}^3$) was obtained from the ratio of the mass of a quantity of wheat kernels to its volume. The volume of the wheat kernels was measured by a Beckman air comparison pycnometer (Model 930, Beckman

Instruments, Inc., Fullerton, CA). The bulk porosity was 0.435, calculated using Eq. 2.

LOCAL AIR VELOCITY AND STATIC PRESSURE IN THE WHEAT BED

The airflow rate supplied to the wheat bed varied from $0.006 \text{ m}^3\text{s}^{-1}\text{m}^{-2}$ to $0.133 \text{ m}^3\text{s}^{-1}\text{m}^{-2}$. The corresponding static pressure measured in the plenum varied from 50 to 1000 Pa. Fig. 4 shows two examples of the air velocity and the pressure measured within the wheat bed at plenum pressures of 400 and 900 Pa. The variations in the pressure readings at each level of the bed were generally less than 5% of the mean value. The measured air velocity did not show a definite trend with grain height. The coefficient of variations (standard deviation divided by the mean) in the measured air velocities was less than 8%. Variations in the measurement could be attributed to sensor accuracy or to variations in the local grain packing.

Fig. 5 shows the average of the measured local air velocity, V_m , at the ten sampling points in the bulk wheat, and the two calculated air velocities, V_{01} and V_{02} . V_{01} and V_{02} are equal to the superficial air velocity, V_0 , divided by the bulk porosity, 0.435. For V_{01} , V_0 was derived from the volumetric airflow measurement. For V_{02} , the measured pressure drop in the plenum was used to estimate V_0 from the data published in ASAE D272.2 (ASAE, 1992). Fig. 5 indicates that V_{01} and V_{02} are in close agreement, as is expected. The air velocity measured within the bulk, V_m , however, was consistently higher than V_{01} and V_{02} . The difference increased with an increase in airflow rate and static pressure.

Fig. 6 is a plot of the ratio of V_0 to V_m versus the static pressure drop per unit depth. The ratio increases from 0.05 at 33 Pa/m to a constant value of 0.14 at about 300-370 Pa/m. When the pressure drop is about 360 Pa/m, the Reynolds number is calculated as 135, which is based on the volume-equivalent particle diameter of the wheat kernels (Li et al. 1991), 3.5 mm, and the measured air velocity at this point. The initial increase in V_0/V_m could be due to a transition from laminar flow to turbulent flow as the pressure drop increase. If the measured air velocity, V_m , is the true local air velocity, the constant value of 0.14 beyond 360 Pa/m may represent the effective fractional open area for airflow in the wheat bed. This value is about 1/3 of the bulk porosity of the grain.

FURTHER TESTS ON WHEAT AND FIELD PEAS

Because wheat kernels are irregular in shape, it was decided to test a more spherical

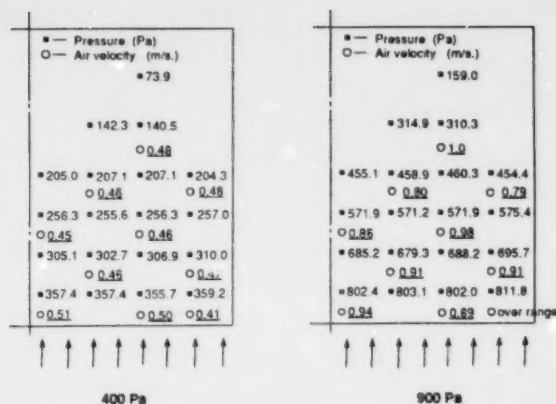


Fig.4 Measured static pressures and air velocities within the wheat bed with 2 plenum pressures

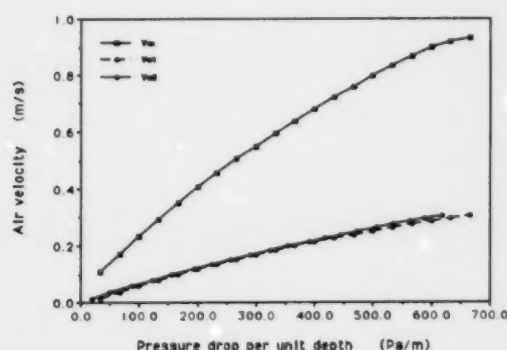


Fig.5 Measured air velocity (V_m) compared to air velocity calculated using Eq. 1 (V_{o1} & V_{o2})

and uniformly sized seed such as the Trapper field pea. The major, intermediate, and minor diameters of one hundred pea kernels were measured with a digital caliper as 5.83 mm (Std. 0.37), 5.61 mm (Std. 0.37) and 5.46 mm (Std. 0.37), respectively. An average sphericity for the pea kernels was calculated as 0.97, using the equation defined by Mohsenin (1986). The average moisture content of the pea samples was 11%. The bulk density was 807 kg/m^3 , and the kernel density was 1403 kg/m^3 . Eq. 2 yields 0.423 for the bulk porosity of the peas

Fig. 7 is a schematic diagram of the equipment used for the tests. Grain kernels were placed in the circular column 320 mm in diameter, with a bed depth of 300 mm. The local air velocity in the bulk grain was measured with two sensors, one with the protective cover and the other without. To minimize possible damage to the uncovered probe, the column was first filled with grain to a depth of 150 mm, the transducers were

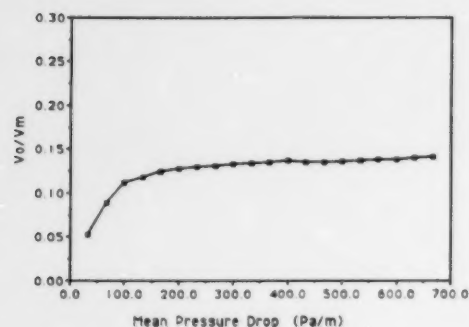


Fig.6 Ratio of V_o to V_m for bulk wheat in experimental bin of Fig.1

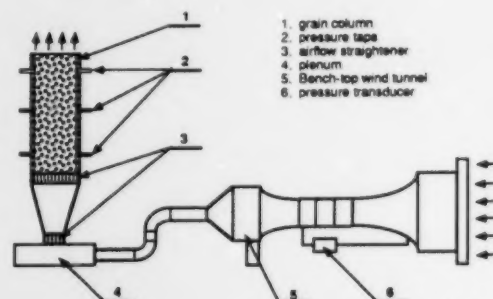


Fig.7 Schematic diagram of experimental equipment with circular bin

laid on the grain, and then covered gently with grain to a total depth of 300 mm.

A preliminary test was performed in the bulk peas. In the test, the supplied airflow was adjusted to a preset rate until a steady airflow was achieved, after which the air velocity in the pea bulk was measured. Fig. 8 shows the output of the covered and uncovered sensors when the superficial air velocity was 0.094 m/s . At time zero, the power supply to the sensors was turned on, followed by a period of 550 to 600 s of recording. The airflow supply was then turned off (sensor remained on) for about 600 s, followed by resuming the airflow supply to the pea column.

The following observations were obtained from the output:

1. Both the covered and uncovered sensors recorded an interstitial velocity proportional to the superficial velocity.
2. The uncovered sensor recorded higher values than the covered one.
3. The output from the sensors did not return to zero upon turning off the airflow supply. The uncovered sensor registered a higher zero drift.
4. The differences in output between the covered and the uncovered sensors remained

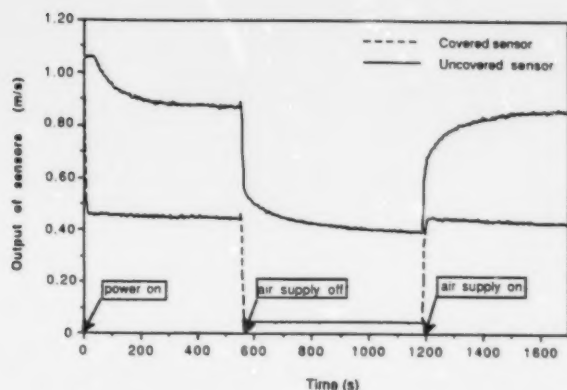


Fig. 8 The output of the sensors placed within the bulk grain peas

almost constant throughout the test.

5. The uncovered sensor had a slower response than the covered one.

Observations 1 to 5 indicate that uncertainties in the measurement of air velocity within the bed using the uncovered sensor could largely be due to the heat conduction and lack of temperature compensation. This is because in the test the uncovered sensor was in touch with the surrounding grain kernels, and the sensor tip was separated by the grain kernels from the temperature compensation points. The covered sensor, on the other hand, did not have conduction and temperature compensation problems because it had no contact with grain kernels, but sensed a "flow" passing by it. This flow is directly proportional to the local air velocity. Considering the dimensions of the cover and the slots on the cover, we expected that the velocity indicated by the covered sensor would be less than the velocity past the grain particles.

More tests were performed in bulk wheat and bulk peas in the same manner as the preliminary test under two superficial air velocities, V_o , 0.05 m/s and 0.094 m/s. Table 1 lists the magnitude of measured air velocities. In Table 1, V_{m1} and V_{m2} , the air velocities measured by the covered and uncovered sensors, are the net values from the outputs of the sensors minus the zero drift values of the sensors. The standard error of the mean given in the parenthesis indicates that the error is less than 2% in the measurements. In all cases the ratio of V_{m1} to V_o or V_{m2} to V_o was higher than 1/ε by a factor of 2 to 3.

The jet velocity of air emerging from the top surface of grain bed was also measured. A spherical sensor (TSI Model 8470) and a needle-like sensor (TSI Model 8450), both

Table 1. Intergranular air velocities (m/s) measured in beds of peas and wheat. $n = 12$

Variety	V_o	V_{m1} (SEM)	V_{m2} (SEM)	V_{m1}/V_o	V_{m2}/V_o
Peas	.050	.303(.005)	.228(.007)	6.06	4.56
	.094	.576(.009)	.475(.014)	6.12	5.05
Wheat	.050	.322(.006)	.316(.003)	6.44	6.32
	.075	.491(.009)	.477(.011)	6.54	6.36

Note:

V_o : the superficial air velocity

V_{m1} : the magnitude of air velocity measured by sensor with the cover

V_{m2} : the magnitude of air velocity measured by sensor without the cover

SEM: Standard error of the mean

Table 2. Air velocity measured on the surface of grain beds

	V_o	V_{s1}	V_{s2}	V_{s1}/V_o	V_{s2}/V_o
On the surface of pea bed	0.037	0.26	0.20	6.89	5.51
	0.050	0.31	0.27	6.28	5.40
	0.063	0.37	0.34	5.89	5.44
	0.075	0.43	0.42	5.67	5.56
	0.094	0.51	0.53	5.38	5.61
On the surface of wheat bed	0.125	0.63	0.72	5.04	5.72
	0.037	0.22	0.19	5.94	5.13
	0.050	0.27	0.25	5.40	5.00
	0.063	0.32	0.31	5.08	4.92
	0.075	0.37	0.38	4.93	5.07
	0.094	0.45	0.50	4.79	5.10
	0.125	0.60	0.70	4.80	5.60

Note:

V_o : the superficial air velocity

V_{s1} : the magnitude of air velocity measured by needle-like sensor

V_{s2} : the magnitude of air velocity measured by spherical sensor

uncovered, were used for the measurement. The sensing element of each of the two air velocity transducers was placed as close as possible to the grain surface without touching any grain kernel. For the test, the superficial air velocity, V_o , ranged from 0.037 m/s to 0.125 m/s. The results of the measurement are summarized in Table 2, where V_{s1} is the magnitude of the air velocity measured by the needle-like sensor, and V_{s2} is the magnitude of the air velocity measured by the spherical sensor. The ratio of V_{s1} to V_o or V_{s2} to V_o seem to be random variables. The ratio of V_{s1} to V_o exhibits a slight decreasing trend with V_o . The ratio of V_{s2} to V_o , on the other hand, shows a slight increase. The reason for these trends may be that in the experiment, maneuvering of the needle-like sensor was limited. Therefore the sensor could not be located exactly at the center of the air jets and so did not register the maximum air velocity consistently. The

spherical sensor has a larger sensor tip so that the effect of exact location of the tip is not significant.

An analysis of variance was performed on group velocity ratios for each grain type. The resulting F-test showed no significant difference in ratios within and between the groups at 95% confidence.

The air velocities measured either on the surface of the wheat bed or on the surface of the trapper field pea bed were more than twice the air velocities calculated from Eq. 1. The results were consistent with those measured earlier within the grain bulk.

DISCUSSION

In studying the flow phenomena in porous media with low and intermediate porosities, conduit flow could be appropriate, that is, the open space available to airflow within grain bulk can be imagined as a bundle of small channels (Carman 1956, Dullien 1975, Sutherland et al. 1971, Brakel 1975, etc.). Based on the conduit flow model, we assume that the cross sectional area of a grain bed is A_0 , and the effective cross sectional area of the small channels for airflow in the bed is A . Then according to the continuity principle for incompressible flow, the following relationship for the air passing through the bed is derived:

$$V_0 A_0 = V A \quad \text{or} \quad V = V_0 / \epsilon_1 \quad (5)$$

where $\epsilon_1 = A/A_0$ is an effective fractional open area. It appears that in most of literature the effective fractional open area, ϵ_1 , is assumed to be equivalent to the bulk porosity of the beds. However, from Carman's work (Eq. 3), ϵ_1 should be equal to the bulk porosity divided by the tortuosity. According to the conclusion drawn by Garret and Brooker (1965) and the results of our experiments, the effective fractional open area, ϵ_1 , of the wheat bed seems to be about one third to two third of the bulk porosity of the bed. It appears therefore that the tortuosity is one of the factors affecting the effective fractional open area, ϵ_1 , and the assertion made by Garret and Brooker that "the actual forward velocity during the fraction of time that the air particles are moving forward must be higher" is a correct one.

Our experimental results indicate that the local air velocity within a grain bulk is under-estimated when Eq. 1 is used. To predict local air velocity within bulk grain more accurately, knowledge of either true bulk porosity or surface porosity of grain bulk is required. Due to the complex structure of the pore spaces in bulk grain, more work is required both theoretically and experimentally to develop a method of arriving at the true

velocity of air within the grain bulk.

CONCLUSIONS

The local air velocity within a grain bulk may be under-estimated by a factor of 2 to 3 times when the air velocity is calculated by dividing the superficial air velocity by the bulk porosity of the bed.

ACKNOWLEDGMENT

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ANAEROBIC DIGESTION AND ENERGY RECOVERY FROM MUNICIPAL AND INDUSTRIAL WASTE SYSTEMS

T. Viraraghavan and P. Tontiwachwuthikul

Faculty of Engineering, University of Regina

Regina, Saskatchewan

S4S 0A2

ABSTRACT

This paper reviews the fundamentals of anaerobic digestion and its application in municipal sludge stabilization and in the treatment of varieties of industrial wastewaters. The paper also discusses biogas recovery and various options available for the utilization of biogas produced in the treatment plant.

INTRODUCTION

The use of biogas from sludge digestion is gaining in importance, because of rising energy costs and the availability of a number of practical and economical systems for such use. Biogas utilization systems such as electricity generation systems may be economically attractive even in smaller treatment plants.

THEORY OF ANAEROBIC DIGESTION

Anaerobic digestion of wastewater sludges/industrial wastes is accomplished by biological conversion of organics to methane and carbon dioxide in an oxygen-free (anaerobic) environment. Figure 1 shows a three-stage scheme involving the following: 1) hydrolysis, liquefaction, and fermentation; 2) hydrogen and acetic acid formation; and 3) methane formation (Parkin and Owen, 1986).

The pathways for methane production in anaerobic digestion are shown in Figure 2. The primary acids produced during acid formation (acetic and propionic) account for the largest percentage of total methane formed. For example, it was found in the case of a complex waste that acetic acid contributed greater than 70% of CH_4 formed and propionic acid was responsible for approximately 10 to 15% of CH_4 produced (McCarty, 1968).

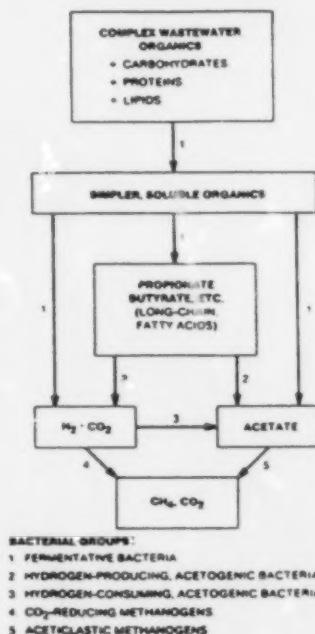


Figure 1. Methane Formation in Anaerobic Digestion (Parkin and Owen, 1986).

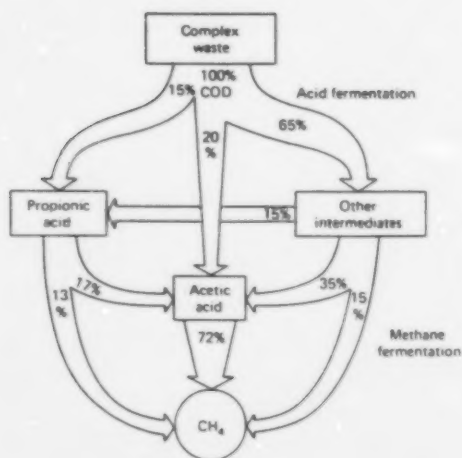


Figure 2. Pathways for Methane Fermentation of Complex Wastes (McCarty, 1968)

BIOGAS - QUANTITY AND QUALITY

The quality of biogas produced during digestion is directly related to the amount of organic matter destroyed. For every kilogram of COD destroyed, regardless of substrate, 0.35 standard cubic metre of methane is produced. In the case of sludge digestion, total gas production is usually estimated from the volatile solids loading of the digester or from the volatile solids reduction. Typical values are from 0.5 to 0.75 m³/kg of volatile solids added and from 0.75 to 1.12 m³/kg of volatile solids destroyed.

A healthy anaerobic digestion process for municipal wastewater sludge produces a biogas with about 65 to 70 percent methane by volume, 25 to 30 percent carbon dioxide and low levels of nitrogen, hydrogen and hydrogen sulfide. Methane content in the biogas can be higher in the case of anaerobic treatment of certain types of industrial wastewaters. The amount of methane which is present in a dissolved phase from an anaerobic reactor treating a dilute wastewater can represent a substantial part of the methane formed in contrast to an anaerobic sludge digester where the high-strength sludge feed make the amount of methane found in solution negligible in comparison to the amount which is recovered in the overlying gas phase. Assuming a mean COD concentration of 300 mg/l for a dilute wastewater, 70% CH₄ in biogas and 70% COD reduction in the anaerobic reactor, the amount of methane lost in the effluent (liquid phase) may be approximately 30 percent at a partial pressure of 760 mm Hg in the case of a reactor operating at 20°C. The energy recovery potential for anaerobic systems operating at low loading rates using low-strength wastewaters is diminished due to the soluble methane loss in the filter effluent. This loss would be higher at lower temperatures, as solubility increases with the decrease in temperature.

One cubic metre of methane at standard temperature and pressure has a heating value of 35800 kJ. The heating value of the biogas containing approximately 60-65% methane is usually taken as 22400 kJ/m³.

BIOGAS UTILIZATION

Options for the use of biogas are listed below:

1. direct burning for space heating and digester heating, cooking, lighting, and for gas-fired refrigeration and air conditioning;
2. drying of sludge;
3. fuel mobile vehicles such as cars, trucks and tractors; and
4. fuel a stationary internal combustion engine driving a generator to produce electricity.

Space Heating

The first option is practised in a number of treatment plants by way of heating buildings and heating the primary digester.

Drying of Sludge

Drying of wastewater sludge by means of biogas is practised in some plants where sufficient gas is available. Generally heat from the gas is used to evaporate the excess water from sludge which has been partly dewatered using, for example, vacuum filters.

In sludge drying, the moisture in the sludge is reduced by evaporation to 8 to 10 percent by application of hot air, without combusting the solid materials. For economic reasons, the moisture content of the sludge must be reduced as much as possible through mechanical means prior to heat drying. The five available heat treating techniques are flash, rotary, toroidal, multiple hearth and atomizing spray. Though heat drying of sludge was developed more than 50 years ago, it is not widely used, as it is quite expensive; it can become viable only when the dried product can be successfully marketed. Approximately 3260 kJ are needed to vaporize one kg of water, based on a thermal efficiency of 72 percent. At times it may be possible to dry sludge to 40-50% moisture content, when the sludge itself has sufficient latent heat to support its own combustion. To accomplish the drying, an auxiliary fuel such as excess gas, if available, or fuel oil could be used.

Use of Biogas in Mobile Vehicles

Biogas can be used as fuel for motor vehicles, but its efficiency is dependent on the methane content, the content of CO₂ and H₂S. Furthermore if enough fuel is to be carried in the vehicle to permit reasonable distance of travel, the gas must be "concentrated" by being compressed and stored in high pressure cylinders. Thus the use of biogas in internal combustion engines of cars, trucks, etc., requires special equipment and processes, which include the following:

1. reducing the H_2S content of the gas to a permissible level to prevent corrosive damage to metal surfaces, particularly bearings and other working parts;
2. a scrubbing system to remove CO_2 (while carbon dioxide exerts no harmful effects on internal combustion engines, the presence of non-combustible gas reduces the heat content per unit of gas volume and thus lowers the operating efficiency of the engine);
3. a compressor capable of compressing gas to pressures between 13800 and 20690 kPa (when gas is to be compressed, it is imperative that carbon dioxide be removed to prevent mechanical damage to compressor caused by liquefaction of the CO_2);
4. high pressure cylinders that can safely store gas at pressures up to 16550 kPa, to be installed on the vehicle;
5. a set of similar high pressure storage cylinders equipped with a pressure control panel for filling the vehicle cylinders;
6. pressure-reducing valves installed on the vehicle to supply the low pressure gas required by the carburetor (usually two valves are used); and
7. an automatic gas/air mixing valve installed on the air intake side of the carburetor.

Conversion kits are commercially available for installation on standard internal combustion engines to permit their operation on either gasoline or methane. The operation of engines using such conversion kits will not provide optimum performance under the option of selecting either fuel. One such conversion kit, installed on a half-ton pickup truck powered by a six-cylinder engine at the University of Manitoba, gave a fuel consumption value of 194 L of methane per kilometre. Successful use in mobile units is contingent upon obtaining adequate concentrated storage to provide a practical distance range. A compromise must be made between pay load and fuel capacity in such applications. Modesto, a city in California, has been using cleaned and compressed methane for fueling five city vehicles suitably modified. Operation of stationary engines should have economic priority over mobile vehicles because there is no need for compressing equipment and high pressure storage tanks. Scrubbing equipment to reduce hydrogen sulfide to a tolerable level, may be necessary as required in both cases to eliminate corrosion of bearings.

Use of Biogas for Power Generation

Biogas can be used for on-site generation of electricity. Diesel or gas internal combustion (IC) engines can be used to drive electric generators.

Diesel engines operate on fuel oil that is ignited entirely by the heat resulting from the compression of the air supplied for

combustion. Gas-diesel engines operate on a combustible gas (sludge gas in this case) as primary fuel; the ignition of the digester gas is accomplished by the injection of a small amount of pilot fuel oil. Generally, 8 to 10 percent fuel oil is required to operate a dual fuel engine. Dual fuel diesel engines are equipped to operate on fuel oil only or as a gas-diesel. Fuel oil is normally used in the alternate fuel system for dual fuel engines in a wastewater treatment plant.

A gas engine is an IC engine that operates on a combustible gas fuel (biogas in this case) that is ignited by an electric spark. Natural gas or propane could be used as an alternative source of fuel in a gas engine; it can be equipped to run on gasoline as well.

There are many variations in engine design and auxiliary equipment, required for these two basic engine types. The operating speed and turbocharging are basic differences between engines supplied by different manufacturers.

It is generally assumed that work can be produced by an IC engine operating on biogas with the efficiency assumed at 35 percent. The efficiency of engines varies depending upon the basic engine design and method of operation. In general, low speed, turbo-charged or dual fuel engines require less fuel per MJ than higher speed naturally aspirated engines. An IC engine/generator's typical efficiency is 25%.

The use of heat recovery equipment will increase the overall efficiency. Typical heat recovery rates in percent of fuel supplied to the engine are: jacket water, 18 to 20%; exhaust 10 to 13%; combination of both jacket water and exhaust heat recovery, 28 to 33%. This recovered heat added to the efficiency of the engine results in a total thermal efficiency ranging between 50 and 70 percent.

It is necessary to examine whether there is a need to clean the gas and store prior to use in an engine for power generation. Wesner and Clarke (1978) noted that even though levels of 1000 to 3000 ppm of H_2S were in the digester gas at several operating plants, no adverse effects had been seen on the engines. The effect of H_2S is significantly reduced or possibly eliminated when exhaust gas is maintained at temperatures greater than its dew point, to prevent condensation. This generally requires that the water temperature of any boiler or engine using unscrubbed gas be at least 100°C. Also stack gas temperatures should not be allowed to drop below 177° to 204°C. Use of unscrubbed gas is preferred. Intermittent operation is not preferred, since condensation will occur each time a unit is shut down.

A number of wastewater treatment plants in U.S.A. and Canada with anaerobic digestion use biogas for power generation (Wesner and Clarke, 1979).

BIOGAS PROCESSING

Removal of impurities from biogas such as H_2S , CO_2 and water is necessary, especially in large-size plants (e.g. more than 1 million ft^3 per day). The reasons for concerns are outlined below:

(i) the H_2S in biogas forms SO_2 after combustion process which then reacts with moisture producing sulfuric acid resulting in environmental and corrosion problems;

(ii) if natural gas pipeline systems are to be used for biogas transportation, the H_2S and CO_2 content must be reduced to be less than 0.006 gm/m^3 and 2 (vol.) %; respectively;

(iii) the CO_2 reacts with the water vapor in the biogas to form a weak acid which attacks metal parts of the biogas utilization equipment;

(iv) the presence of CO_2 reduces the thermal efficiency of biogas utilization equipment; and

(v) at high pressure and low temperature, the CO_2 and water vapor tend to liquefy and form a solid which creates plugging and corrosion problems resulting in high maintenance costs.

Incidentally, the composition of biogas is very similar to that of the natural gas or refinery gas which is mainly methane gas plus some impurities of H_2S and CO_2 . Only major difference is that biogas has much lower pressure at source. In large urban cities which need to treat large amounts of wastewater and where treatment plants are located near natural gas plants or oil

refineries, the biogas could easily be injected into their gas treating unit to remove H_2S , CO_2 and/or water vapor. Figures 3 and 4 show how biogas can be attached into gas processing plants and oil refineries.

The methane product from the biogas can then be marketed as natural gas which is one of the world's most extensive and cleanest energy sources. In addition, the methane can also be used as a feedstock for a variety of chemicals. These include hydrogen, ammonia and synthesis gas, which are the most important basic building blocks for chemical and petrochemical industries. Figure 5 shows the petrochemicals that can be derived from methane.

For the cases where a gas plant or oil refinery is not available nearby, a separate gas treating unit is needed to purify the biogas. Many commercial process technologies are available for removing acid gases and water vapor. The choice of a particular process is affected greatly by the specification required for the product gas, the quantity of gas processed and composition of the raw gas being treated. Gas treating is discussed in detail by Kohl and Riesenfeld (1985) and by Maddox (1983). Brief details of major gas treating technologies which could be used effectively for cleaning the biogas, are included.

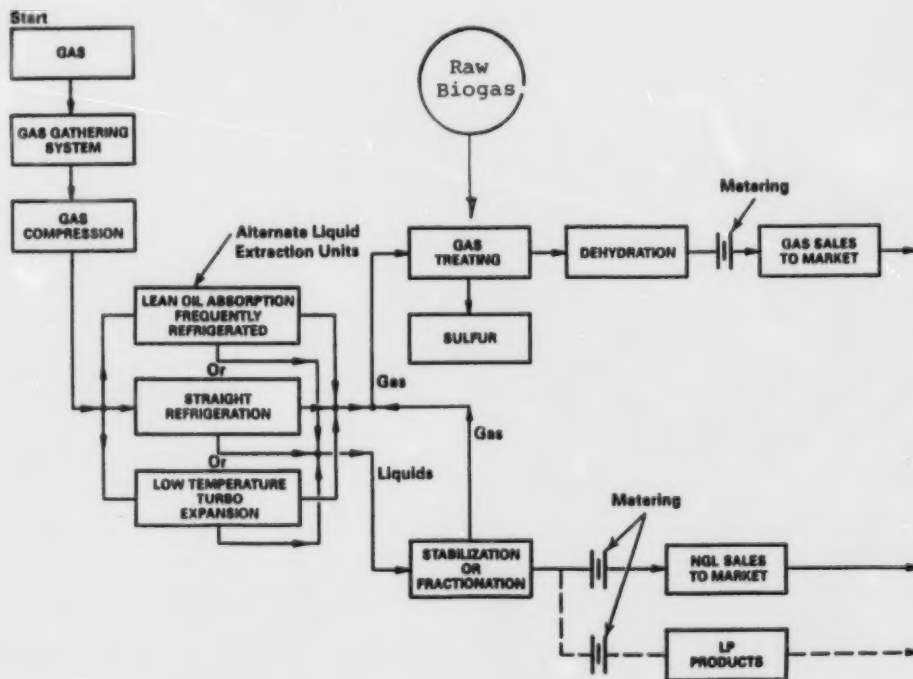


Figure 3: Diagram Showing a Gas Processing System (Katz and Lee, 1990).

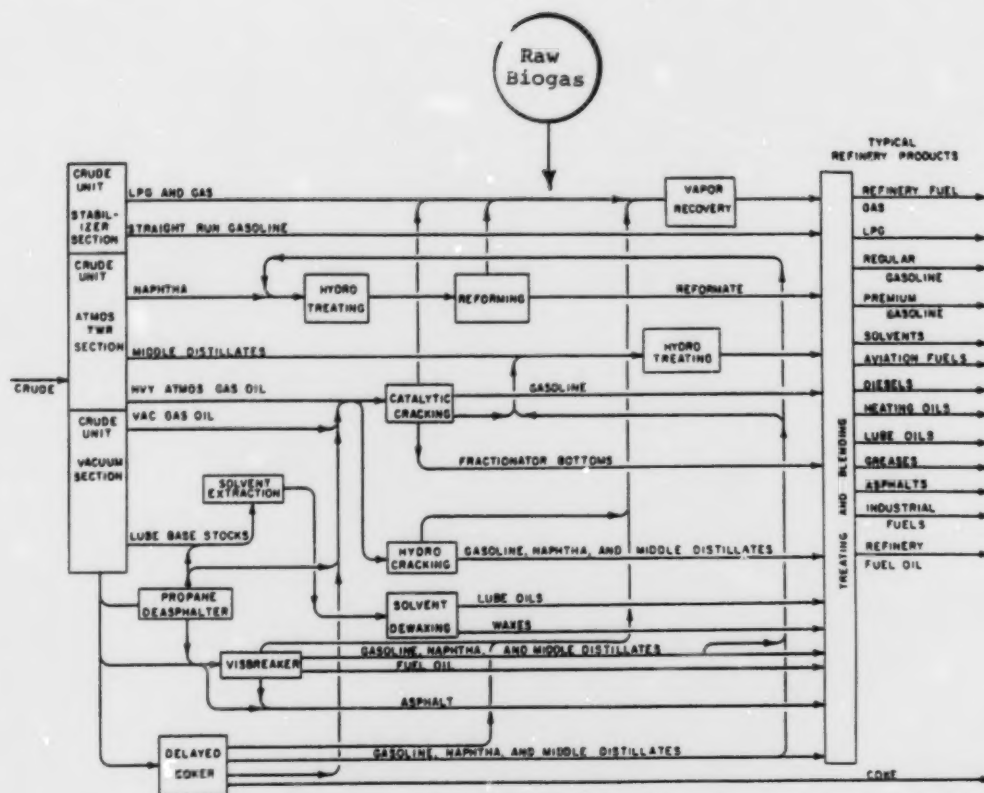


Figure 4: Diagram Showing an Oil Refinery System (Gary and Handwerk, 1984).

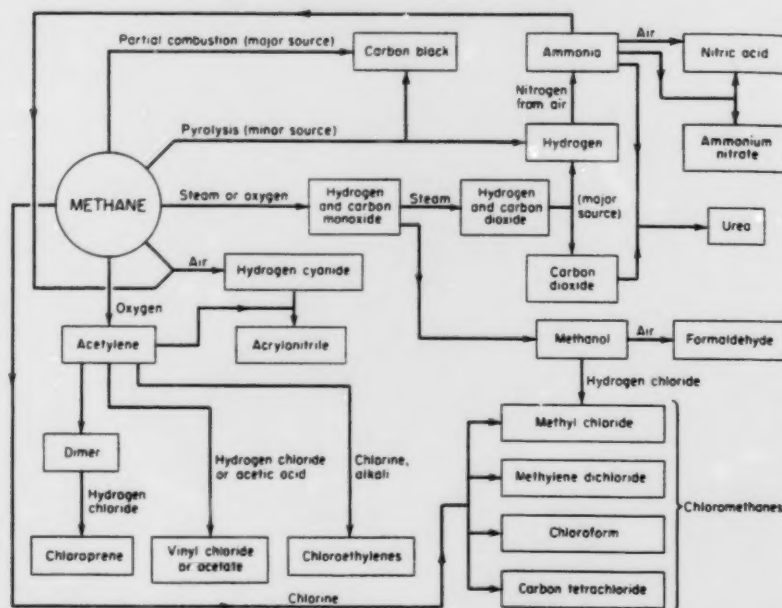


Figure 5: Chemicals and Petrochemicals Derived from Methane (Austin, 1984)

Gas Absorption by Chemical and Physical Organic Solvents

A variety of chemical solvents can be used to absorb H_2S , CO_2 and/or water vapor from biogas. The most commonly used gas treating solvents are alkanolamines such as MEA (monoethanolamine), DEA (diethanolamine), TEA (triethanolamine), and MDEA (methyldiethanolamine). These solvents have been used extensively in natural gas processing plants and oil refineries. Other compounds such as potassium carbonate, sulfolane, propylene carbonate and polyethylene glycol could also be used as treating solvents. A typical flow-sheet of the gas treating with chemical solvent process is shown in Figure 6.

and water vapor. The adsorption process consists of two major steps: adsorption and desorption (regeneration). The technical feasibility of the process is dictated largely by the adsorption step, whereas the desorption step controls its economic viability. Strong affinity of an adsorbent for removing the undesired component from a raw gas mixture is essential for an effective adsorption step. The stronger the adsorptive affinity, however, the more difficult it is to desorb the gas impurity and the higher the energy consumed in regenerating the adsorbent for reuse in the next cycle. The desorption step, therefore, has to be very carefully balanced against the adsorption step for an adsorption process to be successful. This can be achieved by a proper selection of the solid adsorbent.

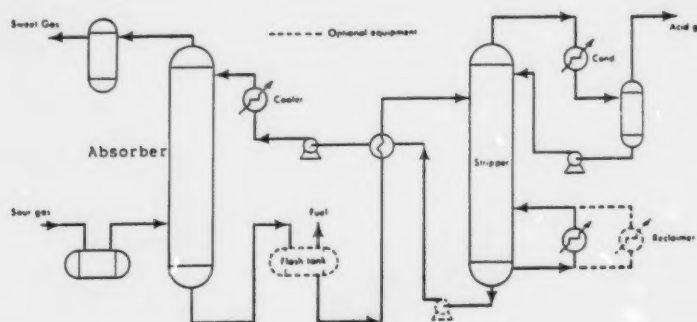


Figure 6: Process Diagram of Gas Treating with Chemical Solvents. (Gas Processors and Suppliers Association, 1983)

The gas containing impurities is contacted counter-currently with the incoming lean solvent in an absorber. The treated gas leaves overhead from the absorber. After the rich (acid gas loaded) solvent leaves overhead from the absorber, it is pre-heated by the hot regenerated lean solvent in a rich/lean exchanger. In the regenerator (or stripper), which is usually a counter-current column, the dissolved acid gas is stripped from the rich solution by steam generated in a reboiler. The lean solvent leaves the regenerator bottom and is recirculated to the absorber top.

For plants of a large scale, this gas treating process is probably the most efficient way for processing the biogas. The technology is well-proven and has been used in gas and oil industries for a long time. However, the amount of gas has to be large in order to operate the process economically and the process involves a substantial capital cost.

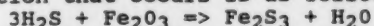
Adsorption by Molecular Sieves

Molecular sieves are solid adsorbents which are specially designed to capture a specific range of gases, including CO₂, H₂S

The gas treating using adsorption process is usually suitable for treating gas with low impurity concentration. In general, the adsorbents can be regenerated by using heat (hot air or steam) to drive off the captured gases.

Dry-bed Chemical Conversion

The gas to be treated is passed through a bed of sponge iron oxide. The chemical reaction that occurs is as follows:



The bed is regenerated by either taking it off line and passing air through it or by continuously injecting a small stream of air with the feed gas. The oxygen in the air reacts with the sulfide and forms elemental sulphur and returns the iron to the oxide. Periodic regeneration is considered much better than continuous operation. This process is normally used to treat only the gas containing less than 1000 ppm of H_2S . The CO_2 in the raw gas is not removed by the iron sponge. In general, the process is suitable for small-scale operations.

Other Processes

In some special situations, other gas treating technologies (e.g. membrane separation, direct reaction with caustics, etc.) could be used to process the biogas. However, the processes discussed earlier, especially the absorption with chemical solvents should be considered first.

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Graphical Simulation of Nonlinear Stresses in Agricultural Food Materials

J. Irudayaraj
Assistant Professor
Agricultural Engineering
University of Saskatchewan
Saskatoon, SASK S7N 0W0

K. Haghighi
Associate Professor
Agricultural Engineering
Purdue University
W.Lafayette, IN 47907

ABSTRACT

The heat and mass transfer process in biomaterials was modeled using a nonlinear coupled partial differential equation and their obtained using the finite element method. The temperature and moisture gradients were calculated and a thermo-hydro viscoelastic boundary value problem was solved to simulate the stresses in the material. Application was made to drying of food grains (barley, soybean and corn). The inner layers of the axisymmetric barley and soybean kernels were in compression while the outer regions were in tension. The graphical software developed in conjunction with the finite element analyser has the capability to animate and display the temperature, moisture and stress distribution and identify the failure regions. The results can be used for grain quality evaluation and establishing optimal drying conditions for crack free drying of cereal grains.

INTRODUCTION

Artificial drying of biological products is one of the most common methods of preservation. However, rapid drying can increase brittleness and induce internal cracks which may cause breakage to occur during subsequent handling. This requires an accurate description of the drying mechanism. Mathematical models that describe grain drying mechanism can provide the required temperature and moisture information for stress analysis. When a porous body dries, expansion occurs during heating and shrinkage occurs because of moisture removal. As a result, highly complicated internal stresses develop because of the initial tendency for the outside of the body to shrink before the interior. An understanding of these drying dependent stresses are therefore essential in order to investigate the efficiency of the existing schedules and to enable the technology to be extended to new products. If the stress state arising from both temperature and moisture gradients can be determined, and if the necessary material properties are known or can be evaluated, a failure criterion can be established which will define the critical temperature and moisture gradients.

Most biomaterials display characteristics of both elastic and viscous fluids and are described as viscoelastic (Morrow and Mohsenin, 1966; Mohsenin, 1984). The exact analytical solution of the governing equations is very difficult, if not impossible to obtain, and therefore approximate solution techniques have to be sought (Muki and Sternberg, 1961). Application of the viscoelastic theory towards studying the behavior of food and biological materials was formulated in the 70's (Hammerle, 1972; Rao et al., 1975). These approaches involved analytical solutions for simple geometries. Recently, the finite element method has been applied to study the

distribution and gradients of temperature and moisture during drying and to simulate the resulting thermo-hydro stresses for a spherical product using constant material properties (Haghighi and Segerlind, 1988). The main objective in this research is to use a general finite element formulation to simulate and study the thermo-hydro stress states and failure phenomenon due to temperature and moisture gradients in viscoelastic biomaterials during drying.

THEORETICAL CONSIDERATIONS

Heat and Mass Transfer Analysis

Finite element formulation and solution of the two nonlinear coupled drying models for two-dimensional and three-dimensional axisymmetric bodies were developed. The Galerkin's weighted residual method was used to develop the FEM formulation (Irudayaraj et al., 1990). This resulted in the matrix equation which has the following form

$$\begin{bmatrix} C_T & 0 \\ 0 & C_M \end{bmatrix} \begin{Bmatrix} \dot{T} \\ \dot{M} \end{Bmatrix} + \begin{bmatrix} K_T & K'_M \\ K'_T & K_M \end{bmatrix} \begin{Bmatrix} T \\ M \end{Bmatrix} - \begin{Bmatrix} F_T \\ F_M \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \quad [1]$$

where C_T , C_M , K_T , K'_T , K_M , K'_M , F_T and F_M are appropriate capacitance, conductance and force coefficients and T and M represent temperature and moisture content, respectively. Nine-noded quadrilateral (Lagrangian) elements were employed and a Gauss quadrature scheme was used to perform the numerical integration for evaluating the element matrices. An explicit three-level and the two-level Crank-Nicolson time stepping schemes were employed to obtain the transient solution of the system of equations.

Thermo-Hydro Viscoelastic Stress Analysis

The simulated temperature and moisture profiles and gradients were then used in a nonlinear viscoelastic boundary value problem to obtain the thermo-hydro viscoelastic stresses developed during drying. The thermo-hydro viscoelastic constitutive equations were first developed considering nonhomogeneous nonconstant temperature and moisture fields. Allowing for the possibility of an initially induced step strain $\epsilon_M(x_i, 0)$, the constitutive equation has the form,

$$\sigma_{ij} = G_{ijM}(\xi) \epsilon_M(x_i, 0) + \int_0^t G_{ijM}(\xi - \xi') \frac{\partial}{\partial \tau} [\epsilon_M(x_i, \tau) - \alpha_M \theta(x_i, \tau) + \beta_M \phi(x_i, \tau)] d\tau \quad [2]$$

Equation [2] indicates that the instantaneous stress at any point in the body is a function of the entire preceding local strain, temperature and moisture history. θ and ϕ are prescribed functions of position and time associated with the solution of nonlinear coupled heat and mass transfer equations (Irudayaraj et al., 1992); $\epsilon_M(t)$ is the infinitesimal strain tensor; $G_{ijM}(t)$ is a fourth order material property tensor; and α_M and β_M are the average thermal and hydro expansion coefficients, respectively.

Invoking the principle of virtual work, which states that at any time instant t , the work done by the internal stresses in domain, D when going through the arbitrary virtual strains $\delta \epsilon_{ij}$ is equal to the work done by the body forces f_i in D when going through the corresponding virtual displacements δu_j , and by the boundary forces $P_j(B)$ when going through the corresponding virtual boundary displacements $\delta u_j(B)$, one obtains

$$\begin{aligned} \int_D (\delta \Delta u_n)^T \{P_n\} dB &= \int_D (\delta \Delta \epsilon_n)^T \{G(\xi_n)\} \{\epsilon_n\} \\ &+ \sum_{m=1}^n \{G(\xi_n - \xi_{m-1})\} \{\Delta \epsilon_m\} dD - \\ \int_D (\delta \Delta \epsilon_n)^T \sum_{m=1}^n \{G(\xi_n - \xi_{m-1})\} \{\Delta(\alpha\theta + \beta\phi)_m\} dD \\ &- \int_D (\delta \Delta u_n)^T \{f_n\} dD \end{aligned} \quad [3]$$

Starting at $t = 0$, the incremental field equations for any time t_m can be written in terms of the past time t_{m-1} and the associated increment Δt_m as

$$t_{(m)} = t_{(m-1)} + \Delta t_{(m)} \quad [4]$$

Applying equations [3] & [4] to the boundary value problem and using the shape function approximation, the general matrix equation for the unknown nodal displacements $\{\Delta \bar{U}_n\}$ at time t_n can be written (Irudayaraj, 1990). From the calculated displacements the elemental strains are obtained from the strain displacement relation and the stresses are then evaluated using (Irudayaraj, 1990)

$$\begin{aligned} \{\sigma_n^i\} &= [G^i(\xi_n)] [B^i]^T [I^i]^{-1} [I^i] [A]^T [C]^T \{\bar{U}_n\} + \sum_{m=1}^n [G^i(\xi_n - \xi_{m-1})] \\ &\{ [B^i]^T [I^i]^{-1} [I^i] [A]^T [C]^T \{\Delta \bar{U}_m\} - \{\Delta(\alpha\theta + \beta\phi)_m^i\} \} \end{aligned} \quad [5]$$

Equation [5] gives the values of the stresses at time $t_{(n)}$ for the I^{th} element. Here, $[A]$ is a boolean matrix representing the connectivity between internal element and externally applied nodal forces, $[B]$ is a matrix of N and its derivatives, $[C]$ is the boundary condition matrix relating the known and unknown displacements. The failure regions within the viscoelastic medium, caused by thermo-hydro effects can be identified from the simulated stress values using a suitable failure theory.

APPLICATIONS

Finite element programs were written to calculate the temperature, moisture and stresses in grain materials during drying. The material properties used in the simulation are as listed by Irudayaraj (1990). Three different types of grains considered for the analysis were barley, soybean, and corn.

The axisymmetric prolate spheroidal model of the barley kernel consisting of 9 nine-noded lagrangian elements, was divided into two regions. Region 1 comprises of elements 2, 3, 4 and 5 and region 2 consists of elements 6, 7, 8 and 9 (Fig. 1). A drying air temperature of 75°C and an initial moisture content of 27% (dry basis) were used to simulate the stresses.

The longitudinal stresses (σ_{xx}) at the innermost region (element 1) were maximum with a value of 3.3 MPa (Fig. 2). Longitudinal stress component σ_{xx} was tensile (with a maximum of 1.1 MPa) at the outer most element and gradually became compressive attaining a maximum value of 3.3 MPa at the inner most element in region 1. As drying proceeds, moisture is removed from the surface of the kernel and the outer layers will contract around a relatively wet incompressible interior. This results in the development of tensile stresses at the outer layers changing to compressive stresses towards the interior. A similar pattern was observed for the simulated stresses in region 2.

Fig. 3 shows a plot of the distribution of stress components in region 1 at 1000 seconds of drying when the longitudinal stress reached its peak (3.3 MPa). At the time of peak longitudinal stress (1000 s) the moisture content in the kernel was 13.0% d.b. This indicates that the stresses due to temperature and moisture gradients are increasing with a steep raise in temperature and fall in moisture content, till the temperature reaches equilibrium. After this time, the rate of increase is reduced, and stresses increase at a slower rate reaching their maximum and then decrease. A similar analysis done on an equivalent spherical model showed a peak stress value of 2.45 MPa at the corresponding radial location. Hence, by approximating a prolate spheroidal grain kernel using a spherical model, one may tend to under predict the compressive stresses developed in the kernel.

The soybean kernel was modelled using a two dimensional axisymmetric finite element grid in cylindrical coordinates consisting of 22 nine-noded Lagrangian elements and 105 nodes. The effect of different drying temperatures on the simulated tangential stress is shown in Fig. 4. Peak stresses observed at the different drying temperatures of 35, 55 and 75°C are 2.8, 3.3 and 3.8 MPa respectively (Fig. 5). Haghighi and Segerlind (1988) observed peak stresses of about 2.1, 4.4 and 6.8 MPa at drying temperatures of 35, 55 and 75°C respectively when, simulating the stresses in a soybean kernel using elastic properties.

Misra et al. (1981) reported a maximum shear stress of about 5.0 MPa in an elastic stress analysis of a soybean kernel and pointed out that the same trend of stresses would be expected for a viscoelastic body, except that the magnitudes of stresses would be increasingly smaller at lower drying temperatures. The magnitude of

the peak tangential stress was smaller at lower temperatures because of the occurrence of stress relaxation. Lower drying temperatures cause smaller thermal and hydal gradients, the internal molecular activity is also slow hence, at such temperatures there is more time available for stress relaxation.

The corn kernel was modelled using 33 nine-noded lagrangian elements and 159 nodes. The finite element grid and the location of the occurrence of simulated maximum stress components, principal and maximum shear stresses for a drying temperature of 85°C, in different regions of the corn kernel are shown in Fig. 5. It should be noted that with the exception of the maximum σ_{xx} in the germ, the occurrence of maximum stress state in the initial stages was at the surface of the kernel, where there is maximum temperature and moisture gradients during the early stages of drying.

GRAPHICAL INTERFACE

A graphical software "GRAINFEM" was developed for the display of temperature, moisture and stress distributions. The program has the flexibility to display a number of parameters. Typically when solving heat transfer, fluid flow or stress analysis problems, the different parameters involved are, temperature, moisture concentration, pressure stress and displacement. Information regarding the various parameters involved at a particular time period might be required to study the state of the material with respect to the unknowns. In this context, multiple object displays on the screen, with each object depicting a variable was chosen to enhance the quality of analysis.

Towards the display of multiple objects, the screen was divided into four quadrants. The user will be given the option of choosing the quadrant for display. Using this option the user can view the status of the material at various times and for various unknowns simultaneously. Further, an option which enables the use of the entire screen is also available.

A display of the temperature or moisture distribution will involve the choice of an appropriate coloring scheme. This was achieved through the implementation of a scan converting algorithm. This algorithm scans and converts a 2-D polygon whose vertices have been mapped to screen (pixel) x,y coordinates. The individual elements in the finite element mesh are treated as polygons whose coordinates and the solution at the nodes are known. Scan converting implies the computation of the corresponding color values of all the pixels that lie inside or on the polygon (Irudayaraj, 1990). An illustration of interpolation using the scan converting algorithm is shown in Fig. 6.

Some of the additional features implemented are zooming, animation, contouring, contour-bands, wire-frame diagrams etc. The zooming algorithm involves clipping procedures. For two-dimensional objects this involves two-dimensional clipping as given by the Cohen-Sutherland (1974) clipping algorithm as discussed by Foley and Van Dam (1982). The clipped region is then displayed by transforming the coordinates from the original location to the quadrant of choice. This transformation can be thought of as a window to viewport mapping, where the window is the original quadrant and the viewport is the chosen quadrant which is magnified proportionally.

The animation option allows the user to observe the behavior of the time varying solution. This is accomplished by displaying the solution dependant color matrix from time = t_0 to time = t_f ,

where t_0 is the initial time and t_f is the final time. Routines were also written for contouring, dynamic contouring, contour banding

etc. All the routines and graphic features were tested and verified in accordance with the results discussed (Figs. 1-5). A flowchart of the GRAINFEM software is shown in Fig. 7.

CONCLUSIONS

An incremental formulation and solution technique is applied to simulate thermo-hydro viscoelastic stresses in grain kernels. The inner layers of the axisymmetric soybean and barley kernels were in compression while the outer regions were in tension. The change of the normal stress components from compressive to tensile is indicative of the fact that the outer layers were shrinking faster than the interior regions of the grain kernel. A spherical approximation of the prolate spheroidal grain kernel (barley) under predicted the maximum stresses developed in the kernel by about 0.85 MPa. This suggests that realistic shape of the product should be used for accurate failure analysis.

For a drying temperature of 85°C, all the stress components reached their maximum at the surface of the corn kernel and at locations where the boundary of the kernel is irregular. The computer graphics program developed in conjunction with the analyzer was very effective in the interpretation of the finite element results and in the identification of potential failure zones.

NOMENCLATURE

$\{F\}$	=	body force vector
$[G(\xi_n)]$	=	material property matrix
M	=	moisture content (%d.b.)
n	=	unit vector normal to boundary
N	=	position dependent function
$P_j(B)$	=	boundary force vector
$\{P_n\}$	=	vector of forces
t	=	time (sec)
T	=	temperature (°C)
u, v	=	element displacement along the x and y axis respectively (m)
$\{U\}$	=	nodal displacement vector

Greek Letters

β_M	=	hydro expansion coefficient($\frac{1}{\%db}$)
α_M	=	thermal expansion coefficient($\frac{1}{^\circ C}$)
θ	=	$T - T_0$
ϵ_{ij}	=	strain tensor
ϕ	=	$M - M_0$
$[\Phi]$	=	matrix of nodal point coordinates
σ_{ij}	=	stress tensor
ξ_i	=	reduced or shifted time (secs)
τ_r	=	relaxation time

Subscripts and Superscripts

1,2,3	=	independent coordinate directions
0	=	base or reference

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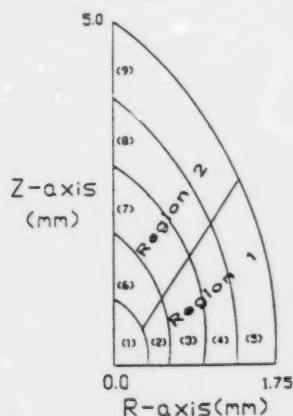


Figure 1. Regions 1 and 2 and corresponding elements of the prolate spheroidal model of the barley kernel.

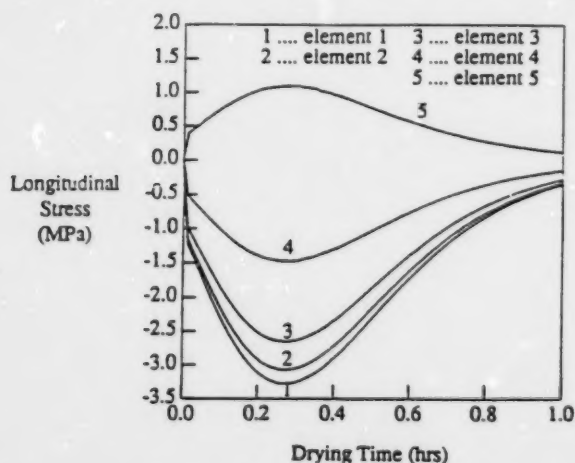


Figure 2. Simulated longitudinal stresses (σ_{zz}) at different locations in region 1 of the prolate spheroidal model of a barley kernel.

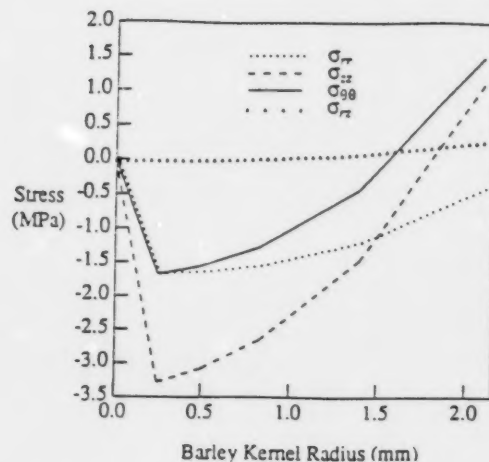


Figure 3. Simulated stresses along region 1 of the prolate spheroidal model at the time σ_{zz} reaches its maximum of 3.3 MPa(1000 s).

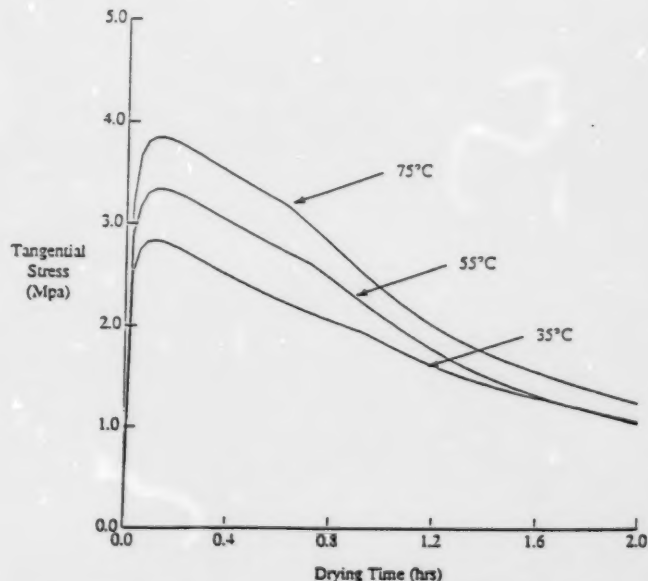


Figure 4. The effect of drying temperature on simulated maximum tangential stress at the surface of a soybean kernel.

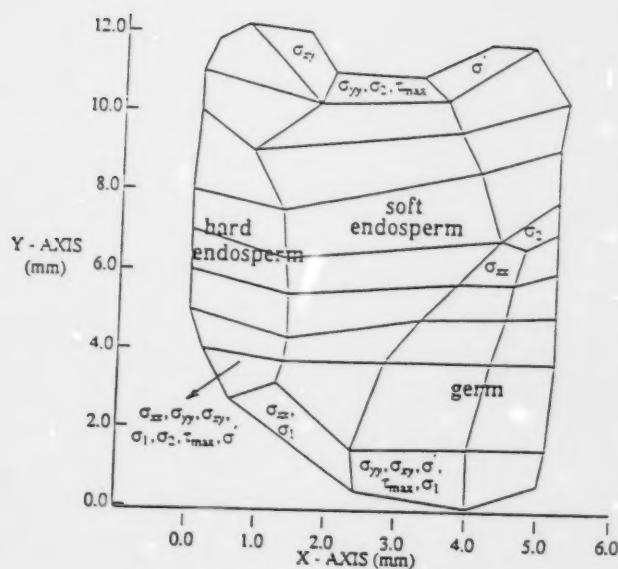


Figure 5. Location of the predicted maximum stress components ($\sigma_{xx}, \sigma_{yy}, \sigma_{zz}$), principal stresses (σ_1, σ_2), maximum shear stress (τ_{max}) and von Mises stress (σ), in a corn kernel during the initial stages of drying (5 minutes).

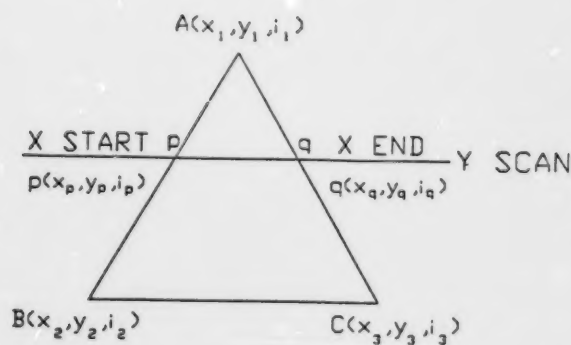


Figure 6. Color intensity interpolation polygon.

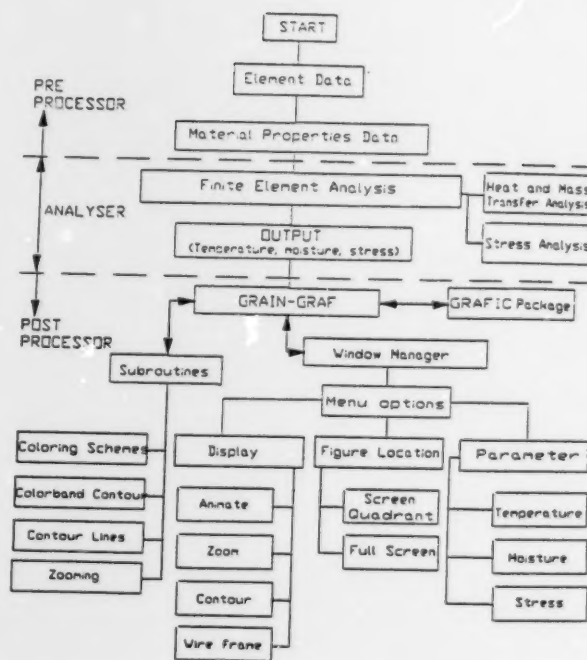


Figure 7. GRAINFEM controls and features.

AN EXPERIMENTAL AND NUMERICAL STUDY OF HEAT TRANSFER IN A SIMULATED COLLECTOR FOR A SOLAR DRYER

P. H. Oosthuizen and A. Sheriff

HEAT TRANSFER LABORATORY
Department of Mechanical Engineering
Queen's University
Kingston, Ontario, Canada

Indirect passive solar crop dryers have the potential to considerably reduce the losses that presently occur during drying of some crops in many parts of the "developing" world. The performance so far achieved with such dryers has, however, not proved to be very satisfactory. If this performance is to be improved it is necessary to have an accurate computer model of such dryers to assist in their design. An important element in any dryer model is an accurate equation for the convective heat transfer in the collector. To assist in the development of such an equation, an experimental and numerical study of the collector heat transfer has been undertaken. In the experimental study, the collector was simulated by a 1m long by 1m wide channel with a gap of 4 cm between the upper and lower surfaces. The lower surface of the channel consisted of an aluminium plate with an electrical heating element, simulating the solar heating, bonded to its lower surface. Air was blown through this channel at a measured rate and the temperature profiles at various points along the channel were measured using a shielded thermocouple probe. Local heat transfer rates were then determined from these measured temperature profiles. In the numerical study, the parabolic forms of the governing equations were solved by a forward-marching finite difference procedure.

NOMENCLATURE

c_p = specific heat
 g = gravitational acceleration
 G = buoyancy parameter, $Gr \sin \alpha / Re$
 Gr = Grashof number
 k = thermal conductivity
 K = constant
 ℓ = length of duct
 L = dimensionless length of duct
 $\overline{Nu}$ = mean Nusselt number based on w
 $\overline{Nu}_\ell$ = mean Nusselt number based on ℓ
 p = pressure
 p_a = atmospheric pressure
 P = dimensionless pressure
 Pr = Prandtl number
 q = mean heat transfer rate from plate
 Re = Reynolds number based on u_m and w
 Re_ℓ = Reynolds number based on ℓ
 T = temperature
 T_a = atmospheric temperature
 T_w = lower plate temperature
 u = velocity component in x-direction
 u_m = mean velocity in duct
 U = dimensionless value of u

v = velocity component in y-direction
 V = dimensionless value of v
 W = width of duct
 x = coordinate along duct
 X = dimensionless value of x
 y = coordinate across duct
 Y = dimensionless value of y
 α = angle of inclination of duct to longitudinal
 β = bulk coefficient
 θ = dimensionless temperature
 ν = kinematic viscosity
 ρ = density

INTRODUCTION

It is widely acknowledged that the introduction of indirect natural convective or passive solar crop dryers in place of traditional drying methods has the potential in many parts of the "developing" world to considerably reduce the losses that presently occur during the drying of some crops such as rice and corn. In such a dryer, the crop is contained within a cabinet in a relatively thin bed which completely spans the cabinet. Air, which is heated in a simple solar collector, then flows, as a

result of the buoyancy forces, up through the crop bed thereby producing the drying. A typical layout of such a dryer is shown in Figure 1.

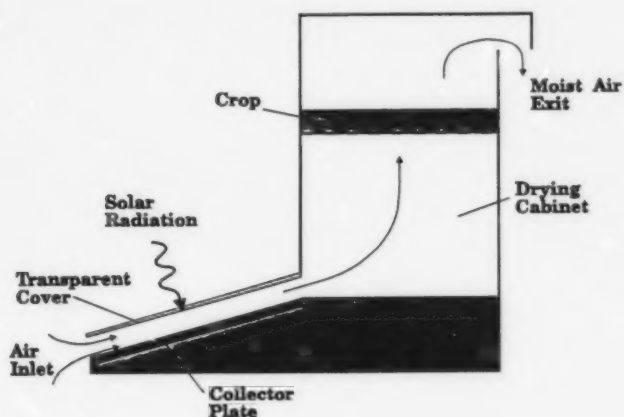


Figure 1 Typical arrangement of an indirect natural convective solar crop dryer

While such indirect natural convection solar crop dryers appear to have wide application in many "developing" countries for the drying of crops such as rice and corn, the performance achieved with such dryers has to date not proved to be very satisfactory. It appears that if this performance is to be improved it will be necessary to have available an accurate computer model of such a dryer in order to produce dryer designs that suit each particular application. A key element of such a model is an accurate equation for the convective heat transfer rate in the collector. Cost considerations usually dictate that a very simple collector be used with such a dryer, this collector usually consisting of a transparent upper surface and a blackened lower surface which is heated by the solar radiation, with the air flowing between the heated plate and the upper surface. The collector is usually inclined at an angle to the horizontal. An accurate equation for the heat transfer rate between the lower solar heated plate and the air flowing through the collector is, therefore, required. In modelling indirect dryers it has frequently been assumed [1] that this heat transfer rate is approximately given by:

$$Nu_L = 1.1 Re_L^{0.5} \quad (1)$$

Field and laboratory testing of dryers has suggested that this equation for the mean heat transfer rate may not always be adequate. To investigate this possibility, heat transfer rates in a simulated collector have, therefore, been measured in the present study for a limited range of conditions. A numerical study of the heat transfer rate has then been undertaken and its predictions shown to compare adequately with the experimental results. The numerical study has then been used to predict the heat transfer for other conditions that are likely to be encountered in actual dryer practice. These extrapolated results have been compared with some rough measurements obtained with actual dryers.

There have been many previous studies of mixed convective heat transfer between parallel plates and in rectangular channels, see for example references [2] to [18]. Much of this work deals with fully developed flow

or with the case where only the temperature field is developing. None of this work appears to have been directed at the range of governing parameters that have been considered here. One important conclusion drawn from this previous work is that it is possible, when the channel is inclined to the vertical, to have a three-dimensional secondary flow develop adjacent to the lower heated plate as a result of the presence of the component of the buoyancy force acting normal to this surface.

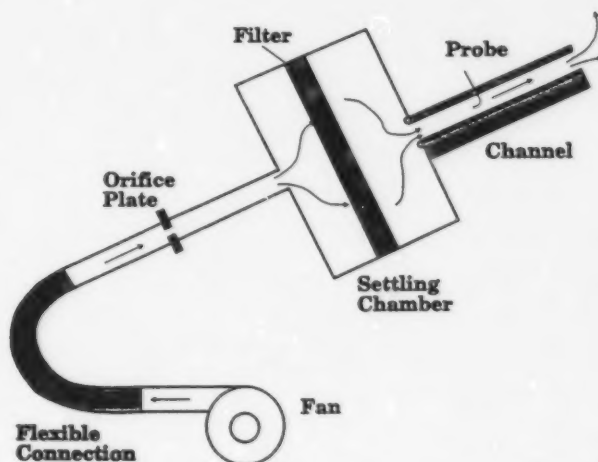


Figure 2 Layout of experimental apparatus

EXPERIMENTAL APPARATUS

The layout of the apparatus used in the present study is shown schematically in Figure 2. The collector was simulated by a 1m long by 1m wide channel with a gap of 4 cm between the upper and lower surfaces of the channel. The lower surface of the channel consisted of an aluminium plate with an electrical heating element, simulating the solar heating, bonded to the lower surface of the plate. This lower heated plate had a series of thermocouples imbedded in it to allow its temperature to be measured. The upper surface of the channel was made from plexiglas. Both the upper and lower surface were well insulated on their outer surfaces. The channel could be inclined at various angles to the horizontal. Air was blown through this channel, using a variable speed fan, at a rate that could be measured using an orifice plate in the pipe connecting the fan to the settling chamber to which the simulated collector was attached. A short rounded inlet was fitted to the channel.

The temperature profiles at various points along the channel could be measured using a shielded thermocouple probe mounted in a traversing gear. Local heat transfer rates were then determined from these measured temperature profiles by using Fourier's law applied to conditions at the heated lower wall.

Initial measurements were undertaken to check that an essentially two-dimensional flow existed in the centre portion of the simulated collector. After this had been established, all further measurements were carried out along the centre-plane of the channel. It should be noted that even if three-dimensional longitudinal rolls exist in the flow for reasons discussed previously, they do not seriously distort the two-dimensionality of the temperature field, although

they can significantly enhance the heat transfer rate. Temperature profiles were measured on the centre-plane at distances of 50mm, 203mm, 356mm, 508mm, 660mm, 812mm and 965mm from the channel inlet plane.

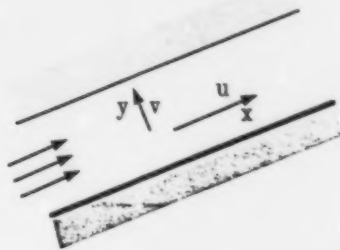


Figure 3 Coordinate system used in numerical analysis

NUMERICAL ANALYSIS

One of the main reasons for undertaking the present experimental study was to investigate the ability of a simple numerical model to predict the heat transfer rate that occurs in the collector. This numerical model is based on the assumption that the flow is laminar and two-dimensional. The fluid properties have also been assumed constant except for the density change with temperature which gives rise to the buoyancy forces, this being treated, therefore, using the Boussinesq approximation. The analysis is based on the use of the boundary layer type or parabolic flow assumptions. Using the coordinate system defined in Figure 3, the governing equations are, therefore, assumed to be:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (2)$$

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = \frac{1}{\rho} \frac{dp}{dx} + \nu \frac{\partial^2 u}{\partial y^2} + \beta g (T - T_a) \sin \alpha \quad (3)$$

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \frac{\nu}{Pr} \frac{\partial^2 T}{\partial y^2} \quad (4)$$

In these equations, the effects of the buoyancy force component in the y-direction has been ignored. This component can cause an instability in the flow leading to a series of longitudinally oriented three-dimensional rolls. The effects of this will be discussed later.

The boundary and initial conditions on eqs. (2) to (4) are:

$$y = 0: u = v = 0, T = T_w$$

$$y = w: u = v = 0, \frac{\partial T}{\partial y} = 0 \quad (5)$$

$$x = 0: u = u_m, T = T_a,$$

$$p = p_a - \rho u_m^2 / 2$$

In writing the initial conditions, it has been assumed that there is a uniform velocity on the inlet plane, that the plate temperature is uniform and that the losses associated with the generation of the initial flow are negligible.

The governing equations have been rewritten in terms of the following dimensionless variables:

$$X = x / w Re, \quad Y = y / w$$

$$U = u / u_m, \quad V = v / u_m \quad (6)$$

$$P = (p - p_a) / \rho u_m^2$$

$$\theta = (T - T_a) / (T_w - T_a)$$

where the Reynolds number, Re , is as usual defined by:

$$Re = u_m w / \nu$$

In terms of these variables, the governing equations become:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (7)$$

$$U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = - \frac{dP}{dX} + \frac{\partial^2 U}{\partial Y^2} + G \theta \quad (8)$$

$$- \frac{dP}{dX} + \frac{\partial^2 U}{\partial Y^2} + G \theta$$

$$U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = \frac{1}{Pr} \frac{\partial^2 \theta}{\partial Y^2} \quad (9)$$

The buoyancy parameter, G , is given by:

$$G = Gr \sin \alpha / Re \quad (10)$$

where the Grashof number, Gr , is, as usual, defined by

$$Gr = \beta g (T_w - T_a) w^3 / \nu^2$$

In terms of the dimensionless variables the boundary and initial conditions become:

$$Y = 0 \text{ and } Y = 1: U = V = 0$$

$$X = 0: U = 1, P = -0.5 \quad (11)$$

$$Y = 0: \theta = 1$$

$$Y = 1: \frac{\partial \theta}{\partial Y} = 0$$

Equations (7), (8) and (9) are the governing equations. They have been solved, subject to the boundary and initial conditions given in eq. (11), using a forward-marching explicit finite-difference procedure. Central differences were used in the Y-direction and forward differences in the X-direction. The mean Nusselt number has been obtained from the numerical results by considering the net enthalpy gain of the fluid, i.e. by noting that:

$$q_L = \int_0^w \rho u c_p (T - T_a) dy \quad (12)$$

which can be rearranged to give:

$$\overline{Nu} = \frac{Pr}{L} \int_0^1 \theta dy \quad (13)$$

where the Nusselt number, $\overline{Nu}$, is given by:

$$\overline{Nu} = q w / (T_L - T_a) k \quad (14)$$

RESULTS

Measurements were undertaken at flow rates that gave Reynolds numbers based on channel height, of between 1000 and 3000 for angles of inclination to horizontal of between 0 and 30 degrees for a lower wall temperature of approximately 80 C. These are conditions

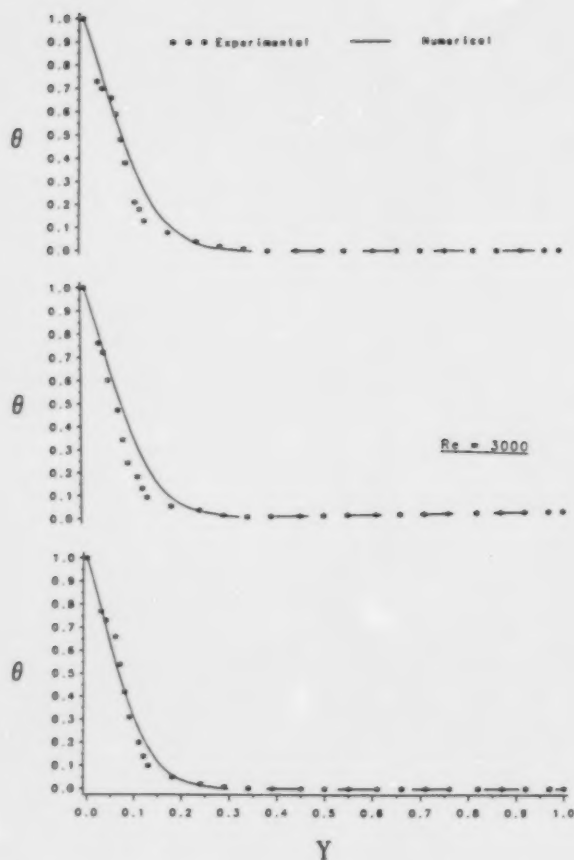


Figure 4 Variation of θ across the channel at a distance of 508 mm from the inlet for a Reynolds number of 3000 for angles of inclination of, from top, 10, 20 and 30 degrees to the horizontal.

that are known to be similar to those that occur in actual drying practice. For the lower plate temperature and gap size used, the Grashof number based on the channel gap and on the difference between the plate and the ambient temperatures is approximately 200,000. While results were obtained for a number of Reynolds numbers, results will only be presented here for Reynolds numbers of 1500 and 3000.

The assumed governing equations for the numerical analysis have as parameters:

- the buoyancy parameter, $G = Gr \sin \alpha / Re$
- the dimensionless duct length, $L = l / w Re$
- the Prandtl number, Pr

Solutions have been obtained only for $Pr = 0.7$, the approximate value for air at ambient temperature. Numerical results have been obtained for the same values of the governing parameters as those existing in the experimental study. In addition, results have also been obtained for the same Grashof number but for lower Reynolds numbers, these conditions being similar to those existing in some experimental studies of the performance of a simulated dryer (see below). In all the calculations, therefore, the Grashof number was assumed to be the same but the dimensionless buoyancy parameter and collector length varied with Reynolds number.

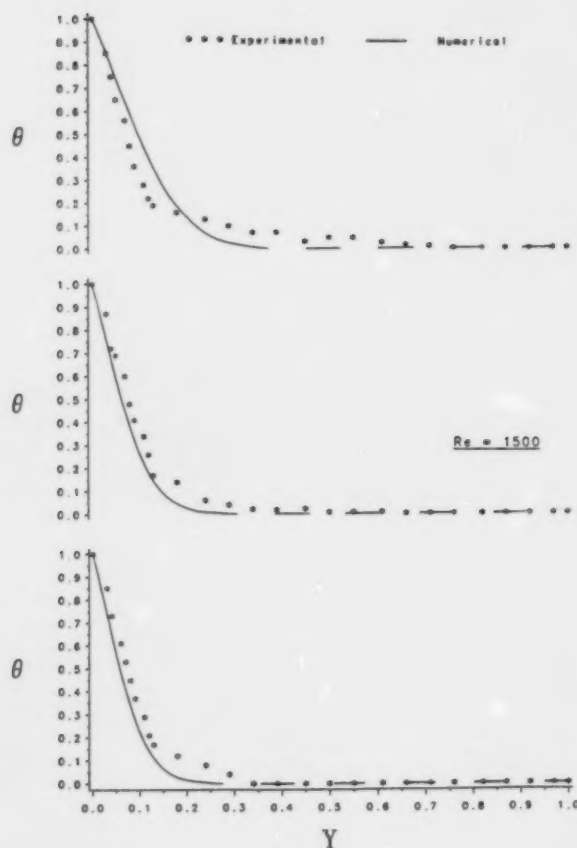


Figure 5 Variation of θ across the channel at a distance of 508 mm from the inlet for a Reynolds number of 1500 for angles of inclination of, from top, 0, 20 and 30 degrees to the horizontal.

Figures 4 and 5 show some typical temperature profiles at approximately half-way along the collector. Both experimental and numerically predicted results are shown in these figures. Quite good agreement between the measured and predicted results will be seen to exist.

Typical local Nusselt number variations along the channel are shown in Figures 6 and 7, both experimental and numerical results again being shown. From these figures it will be seen that the measured variation is quite well predicted except near the channel entrance. The poor agreement in this region is to be expected because:

- the plate temperature was not truly uniform but decreased towards the inlet because of the high heat transfer rates existing in this region i.e. the plate temperature was somewhat lower than the average near the channel inlet;
- the parabolic form of the governing equations do not apply near the inlet;
- the velocity was not truly uniform across the inlet;
- there was some upstream conduction of heat near the channel inlet with the result that the inlet temperature profile was not uniform; and
- the thermal boundary layer is very thin near the inlet with the result that the experimental values have their highest uncertainty in this region.

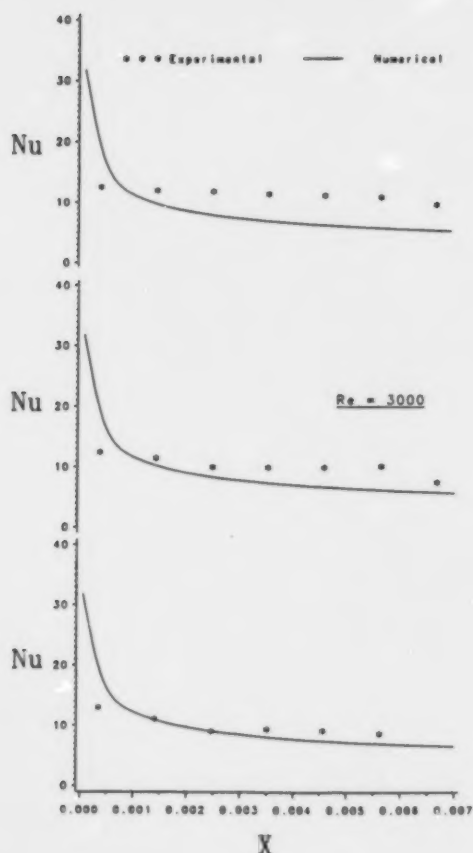


Figure 6 Variation of Nu along the channel for a Reynolds number of 3000 for angles of inclination of, from top, 10, 20 and 30 degrees to the horizontal.

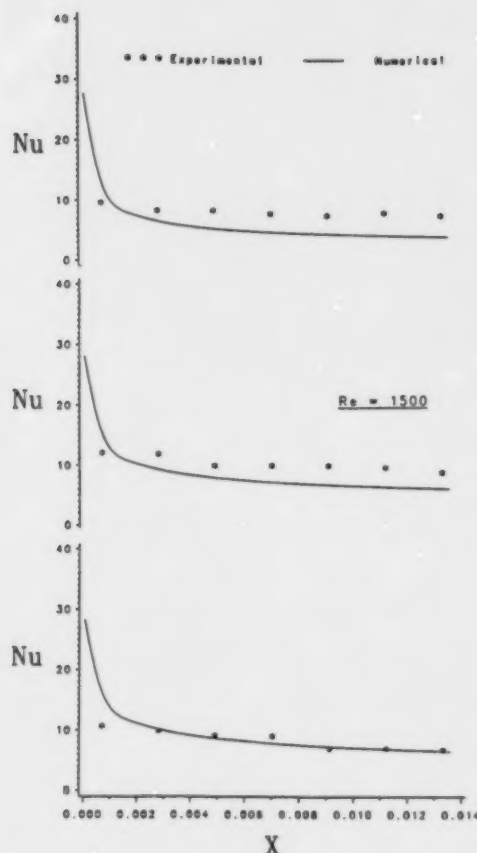


Figure 7 Variation of Nu along the channel for a Reynolds number of 1500 for angles of inclination of, from top, 0, 20 and 30 degrees to the horizontal.

It will also be noted from the results given in Figures 6 and 7 that the experimental and numerical results show the best agreement at the higher angles of inclination. In fact, if the results for an angle of inclination of 30 degrees are considered, it will be seen that if except for the experimental result obtained very close to the inlet, the agreement between the numerical and experimental results is excellent. Since the flow is assumed to be two-dimensional in the numerical analysis and since the importance of the normal component of the buoyancy force decreases with increasing angle of inclination, this strongly suggests that the secondary flow is having a significant influence on the heat transfer rate at angles of inclination of less than about 20 degrees. Since even near the equator, the collector on a dryer will be inclined at least at this angle, this indicates that the numerical analysis gives a good description of the flow and heat transfer in the collector of most solar crop dryers. It should be noted that if the results reviewed in [18] for the conditions under which a secondary flow will develop are considered, they support this conclusion for the conditions covered in the present

tests. It will also be noted from the results given in Figures 6 and 7 that the secondary flow appears to increase the heat transfer rate, a conclusion that has been reached in previous studies.

The measured local heat transfer rates have been integrated to give the mean heat transfer rates in the channel. The results so obtained for an angle of inclination of 20 degrees have been compared with the predicted results for this angle in Figure 8. Also shown in this figure are some results for lower Reynolds numbers deduced from drying rates measured in a simulated dryer, this dryer using an electrically heated plate to simulate the solar heated plate in a real dryer. The uncertainty in these results is relatively large. Nevertheless, it will be seen from Figure 8 that the numerical results are in relatively good agreement with all the experimental results and that the mean heat transfer rate in a collector operating under the conditions considered in the present study can be approximately described by the following equation:

$$\overline{Nu}_L / Re_L^{0.5} = 1.2 (1 + 2350 / Re_L^{1.4}) \quad (15)$$

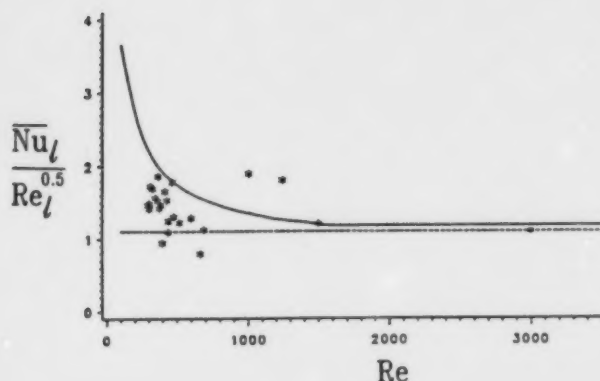


Figure 8 Variation of $\overline{Nu}_L / Re_L^{0.5}$ with Reynolds number. The experimental results for Reynolds numbers of 1500 and 3000 are from the present study. The remaining experimental results have been obtained from drying rates measured in a simulated dryer.

CONCLUSIONS

The results obtained in the present study indicate that a simple two-dimensional parabolic numerical solution gives a good prediction of the heat transfer rate in a dryer collector provided that the angle of inclination of the collector is more than 20 degrees.

ACKNOWLEDGEMENTS

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Heat and mass transfer in shrinking grains

M.J. Miketinac

Department of Mathematics
University of Saskatchewan

and

S. Sokhansanj

Department of Agricultural Engineering
University of Saskatchewan

Abstract

A model simulating the process of simultaneous heat and mass transfer in the drying of a shrinking kernel of barley is formulated. The shrinking is accounted for by a (new) convective transport term in the partial differential equation for the moisture content. The finite element method is used to numerically solve the nonlinear coupled system of two partial differential equations modeling the process of drying with shrinking. Two (new) parameters that appear in the model are determined from published experimental measurements.

1. Introduction

The problem of describing the drying rate of a thin layer of grain (as an important building block of a model that simulates the drying process of a deep bed of grain, Brooker et al., 1974) has attracted considerable attention in experimental and theoretical studies. Miketinac et al. (1992) reviewed five different models of coupled heat and moisture transfer within a single kernel of barley. All the models consist of two coupled nonlinear partial differential equations of parabolic type for temperature, T , and moisture content, M ; additional coupling is introduced by the boundary conditions. The effects of moisture removal on shrinkage has not been studied extensively, but there is a paper by Misra and Young (1980), in which shrinkage of soybeans during isothermal drying is considered.

In this paper a model of simultaneous heat and mass transfer within a single kernel accompanied by shrinking of the kernel is formulated. The model is an elaboration of the approach taken by Aregba et al. (1990) in their study of shrinking gels. Their approach is a simplification of the theory developed by Crapiste et al. (1988). A numerical solution of the model is obtained using the finite element method, which takes an accurate account of nonlinearities and couplings present in the differential equations and the boundary conditions; the fact that the solution must be found in a changing domain is also taken into consideration. The coefficient of cubical shrinkage (Tang, 1991) is obtained from an experimentally found function (Nelson, 1986). The experimental data of Bruce (1985) are used to determine the value of another unknown parameter of the model by applying the inverse method.

2. Heat and mass transfer model of drying of a shrinking grain kernel

Shrinking properties of grains are often characterized by specifying the ratio of volumes (Aregba et al., 1990):

$$S_V = \frac{V}{V_0}$$

as a function of the average moisture content $\bar{M}$. In this equation V_0 is the initial volume, while V is the volume of the grain with the average moisture content $\bar{M}$, where by definition

$$\bar{M} = \frac{1}{V} \int_V M dV.$$

Other descriptions of shrinkage properties of grains are possible. Tang (1991) defines the cubical shrinkage coefficient, β_V , by

$$\beta_V = \frac{dV}{dM}.$$

If β_V is constant, integration of this equation leads to:

$$V = V_0 e^{-\beta_V(M_0 - \bar{M})}.$$

And this shows that:

$$S_V = e^{-\beta_V(M_0 - \bar{M})}. \quad (1)$$

The mass, m_s , of the solid component of the grain remains constant during drying, but its density, as defined by

$$\rho_s = \frac{m_s}{V},$$

changes. Since

$$m_s = \rho_{s,t=0} V_0 = \rho_s V,$$

the function S_V can be written as

$$S_V = \frac{\rho_{s,t=0}}{\rho_s} \quad (2)$$

In the paper by Nelson (1986), there is an experimentally determined expression for ρ_s of barley as a function of the moisture content. That expression is:

$$\rho_s = 1305.9 - 1170m + 4000m^2$$

where m is moisture content (wet basis). The above expression is used in this paper to determine the value of β_V . However, either expression (1) or (2) for S_V can be used in the analysis that follows.

If ρ_l denotes the density of the liquid phase (water), then the conservation of mass equations are

$$\frac{\partial \rho_s}{\partial t} + \text{div}(\rho_s \vec{v}_s) = 0 \quad (3)$$

$$\frac{\partial \rho_l}{\partial t} + \text{div}(\rho_l \vec{v}_l) = 0, \quad (4)$$

where $\vec{v}_l$ and $\vec{v}_s$ are the velocities of the liquid phase and the solid phase, respectively. These two equations can be combined into an equation for the moisture content M (dry basis) defined by

$$M = \frac{\rho_l}{\rho_s} \quad (5)$$

To see this, denote by

$$\vec{J} = \rho_l(\vec{v}_l - \vec{v}_s)$$

as the flux of the liquid phase with respect to the frame of reference that is fixed with respect to the solid phase. Then it can be assumed that

$$\vec{J} = -D \text{grad} \rho_l, \quad (6)$$

where D is the mass diffusion coefficient.

From equation (5) it follows by using (3) and (4) that

$$\frac{\partial M}{\partial t} = \frac{1}{\rho_s} \frac{\partial \rho_l}{\partial t} - \frac{\rho_l}{\rho_s^2} \frac{\partial \rho_s}{\partial t}$$

and

$$\rho_s \frac{\partial M}{\partial t} = \frac{\partial \rho_l}{\partial t} - M \frac{\partial \rho_s}{\partial t}$$

$$= -\text{div}(\rho_l \vec{v}_l) + M \text{div}(\rho_s \vec{v}_s) \\ = -\text{div}[\rho_l(\vec{v}_l - \vec{v}_s)] - \text{div}(\rho_l \vec{v}_s) + M \text{div}(\rho_s \vec{v}_s).$$

By (6) this can be written as

$$\rho_s \frac{\partial M}{\partial t} = \text{div}(D \text{grad} \rho_l) - \text{div}(\rho_l \vec{v}_s) + M \text{div}(\rho_s \vec{v}_s) \\ = \text{div}[D \text{grad}(\rho_s M)] - \text{div}(M \rho_s \vec{v}_s) + M \text{div}(\rho_s \vec{v}_s).$$

The last two terms combine into $-\rho_s \vec{v}_s \cdot \nabla M$ by using the fact that

$$\text{div}(M \rho_s \vec{v}_s) = M \text{div}(\rho_s \vec{v}_s) + \rho_s \vec{v}_s \cdot \nabla M.$$

Hence,

$$\rho_s \frac{\partial M}{\partial t} = \text{div}[D \text{grad}(\rho_s M)] - \rho_s \vec{v}_s \cdot \text{grad} M.$$

This can be simplified by assuming that $\text{grad} \rho_s = 0$, i.e. a uniform distribution of the solid phase. The final equation for M is then

$$\frac{\partial M}{\partial t} + \vec{v}_s \cdot \text{grad} M = \text{div}(D \text{grad} M). \quad (7)$$

This equation for the moisture content contains a new term, i.e. $\vec{v}_s \cdot \text{grad} M$ (see Miketinac et al., 1992), accounting for the shrinkage.

The shrinking velocity, $\vec{v}_s$, can be obtained by integrating (3) under the assumption that shrinking is isotropic and uni-dimensional, i.e. the grain remains spherical. Equation (3) in spherical coordinates has the following form

$$\frac{\partial \rho_s}{\partial t} + \frac{1}{r^2} \frac{\partial}{\partial r}(r^2 \rho_s v_r) = 0,$$

where $\vec{v}_s = v_r \vec{e}_r$, and so by integrating this one obtains

$$r^2 \rho_s v_r = - \int_0^r x^2 \frac{\partial \rho_s}{\partial t} dx$$

or

$$v_r = - \frac{1}{r^2 \rho_s} \int_0^r x^2 \frac{\partial \rho_s}{\partial t} dx.$$

But from (2)

$$\rho_s = \frac{\rho_{s,t=0}}{S_V}$$

and so

$$\frac{\partial \rho_s}{\partial t} = - \frac{\rho_{s,t=0}}{S_V^2} \frac{d S_V}{d t}.$$

Hence

$$v_r = - \frac{1}{r^2 \rho_s} \rho_{s,t=0} \int_0^r x^2 \frac{1}{S_V^2} \frac{d S_V}{d t} dx \\ = \frac{1}{3} r \frac{1}{S_V} \frac{d S_V}{d t}$$

as $S_V, \frac{d S_V}{d t}$ do not depend on x . This means that

$$\vec{v}_s = \frac{1}{3} r \frac{1}{S_V} \frac{d S_V}{d t} \vec{r}. \quad (8)$$

Hence, the differential equations of the model are (7) with (8) and

$$\rho C \frac{\partial T}{\partial t} = \text{div}(K \text{grad} T), \quad (9)$$

where T is the grain temperature and C and K are known functions of T and M .

The boundary conditions for (7) and (9) are

$$-D \frac{\partial M}{\partial n} = h_m(M - M_a), \quad (10)$$

$$-K \frac{\partial T}{\partial n} = h_T(T - T_a) - \rho[h_{fg} + C_V(T_a - T)]h_m(M - M_a), \quad (11)$$

where $\frac{\partial}{\partial n}$ stands for the differentiation in the direction of the outward facing normal, h_T, h_m are the heat and mass transfer coefficients, T_a is the temperature of the drying air, M_a is the

equilibrium moisture content and C_V, h_{f_s} are experimentally determined constants. The boundary condition (11) is justified by Miketinac et al. (1992); the same boundary condition is used by Aregba et al. (1990) in their study of shrinking gels.

The differential system (7),(9),(10) and (11) is completed by giving initial conditions

$$M|_{t=0} = M_0 \quad (12)$$

$$T|_{t=0} = T_0, \quad (13)$$

where M_0 and T_0 are constants.

3. Numerical solution

One of the difficulties associated with numerically solving the system of equations (7)-(13) is the fact that the kernel - assumed to be a sphere for simplicity - shrinks as it dries. There are several ways of handling this difficulty. Aregba et al. (1990) use a moving grid that follows the movement of the solid phase. Kechau and Roques (1990) use a transformation to spatial coordinates moving with the solid phase. In this paper the following transformation of independent variables (Miketinac et al., 1992)

$$x = \frac{r}{R}$$

and

$$t' = \left(\frac{D_0}{R^2}\right)t$$

where $R = R(t)$ is the radius of the sphere representing the kernel and D_0 is the typical value of the diffusivity, D , achieves two purposes. The variables x, t' become dimensionless and x becomes fixed in the interval between 0 and 1.

To transform the differential equations (7) and (9) it is necessary to note that (by the chain rule for derivatives)

$$\frac{\partial}{\partial t} = -\frac{r}{R^2} \frac{dR}{dt} \frac{\partial}{\partial x} + \left(\frac{D_0}{R^2} - 2\frac{D_0}{R^3} \frac{dR}{dt} t'\right) \frac{\partial}{\partial t'}$$

$$\frac{\partial}{\partial r} = \frac{1}{R} \frac{\partial}{\partial x}$$

Using these equations, (7) and (9) become

$$x^2 B \frac{\partial M}{\partial t'} = \frac{\partial}{\partial x} \left(x^2 \frac{D}{D_0} \frac{\partial M}{\partial x} \right) \quad (14)$$

and

$$x^2 \rho C B \frac{\partial T}{\partial t'} = \rho C x^2 G \frac{\partial T}{\partial x} + \frac{\partial}{\partial x} \left(x^2 \frac{K}{D_0} \frac{\partial T}{\partial x} \right), \quad (15)$$

where

$$G = \frac{R}{D_0} \frac{dR}{dt} \quad (16)$$

and

$$B = 1 - 2t'G. \quad (17)$$

The function G in (16) can be rewritten in a more convenient way as follows. Since

$$S_V = \frac{V}{V_0} = \frac{R^3}{R_0^3},$$

it follows that

$$\frac{1}{R} \frac{dR}{dt} = \frac{1}{3} \frac{1}{S_V} \frac{dS_V}{dt}$$

(which can be computed from the experimentally determined function S_V). When differentiating S_V it is necessary to obtain an expression for a derivative of $\bar{M}$. The derivative of $\bar{M}$ is computed from

$$\frac{d\bar{M}}{dt} = -\frac{A}{V} h_m (\bar{M} - M_e),$$

which is derived by Miketinac et al. (1992); in this relation A is the surface area of the kernel. As a result of this

$$G = -\frac{R}{D_0} \left(\frac{1}{S_V} \frac{dS_V}{d\bar{M}} \right) h_m (\bar{M} - M_e). \quad (18)$$

If Tang's expression is used for S_V , i.e. $S_V = e^{-\beta_V(M_0 - \bar{M})}$, then

$$\frac{1}{S_V} \frac{dS_V}{d\bar{M}} = \beta_V.$$

When solving equations (14) and (15) with (17) and (18), the average moisture $\bar{M}$ can be computed by numerical integration and the radius is then obtained from

$$\frac{R}{R_0} = S_V^{\frac{1}{3}}.$$

The boundary conditions at $x = 1$ are

$$-\frac{D}{D_0} \frac{\partial M}{\partial x} = h_m \frac{R}{D_0} (M - M_e), \quad (19)$$

and

$$-\frac{K}{D_0} \frac{\partial T}{\partial x} = h_T \frac{R}{D_0} (T - T_a) - \rho [h_{f_s} + C_V (T_a - T)] h_m \frac{R}{D_0} (M - M_e). \quad (20)$$

It is not possible to construct an exact solution of the initial boundary-value problem for a system of two coupled nonlinear parabolic partial differential equations such as (14), (15), (17) - (20) appearing in the model. In this paper, a numerical solution is obtained by using the finite element method in the form of a package PDE/PROTRAN (marketed by IMSL, 1986). A detailed description of the package is given in two books written by its developer (Sewell, 1985 and Sewell, 1988; see also Miketinac et al, 1987). The package offers much versatility in dealing with nonlinear time-dependent problems such as the one presented by (14) and (15) and the associated boundary and initial conditions. For example, kernel shapes more complicated than the sphere can be easily processed by the package. But the main reason for using the package is that it frees a modeler from a very time-consuming task of implementing the finite element method on a computer.

The model equations contain at least one constant parame-

ter, say α , which is not known experimentally and so the problem considered in this paper is to find α (and possibly other constant parameters), which minimizes the deviation from experimentally measured values T_{exp}, M_{exp} (published by Bruce, 1985); the deviation is defined by

$$J(\alpha) = \sum_{j=1}^n \left[\left(\frac{T_j - T_{exp,j}}{T_j} \right)^2 + \left(\frac{M_j - M_{exp,j}}{M_j} \right)^2 \right]. \quad (21)$$

A very similar deviation is used by Misra and Young (1980); they define

$$j(\alpha) = \sum_{j=1}^n \left[\frac{(T_j - T_{exp,j})^2}{T_{ave}^2} + \frac{(M_j - M_{exp,j})^2}{M_{ave}^2} \right], \quad (22)$$

where

$$T_{ave} = \frac{1}{n} \sum_{j=1}^n T_{exp,j}$$

$$M_{ave} = \frac{1}{n} \sum_{j=1}^n M_{exp,j}.$$

In these definitions T_j, M_j are computed surface temperature and average moisture content values for given α and n is the number of computed values. $T_{exp,j}$ and $M_{exp,j}$ are obtained by a cubic spline smoothing (using IMSL subroutine CSSCV; see IMSL, 1987) of experimental measurements and evaluated at times selected by PDE/PROTRAN.

The problem of minimizing $J(\alpha)$ or $j(\alpha)$ is solved by a modified

Levenberg-Marquart algorithm as implemented in subroutine LMSTR of the package MINPACK (More et al., 1980). This subroutine calls the PDE/PROTRAN package to produce T_j, M_j for $j = 1, 2, \dots, n$ whenever α is set. Expression (22) is a simpler function of the parameter(s) and the execution times, when it is used instead of (21), are faster. Expression (22) is used basically to check the results arrived at by using (21).

4. Results and Discussion

Unfortunately, Aregba et al. (1990) do not provide the complete set of data for the two gels they studied, and so it was not possible to reconstruct their results for the purpose of checking the methods developed in this paper. Instead the published data for barley are used to predict and examine some of its properties that are relevant to shrinking.

For the kernel of barley, which is studied in this paper, $2R_0 = 0.004 \text{ m}$, $\rho = 720 \text{ kg m}^{-3}$, $K = 0.14 + 0.68M \text{ W m}^{-1} \text{ K}^{-1}$, $C = 1300 + 4186.8M \text{ J kg}^{-1} \text{ K}^{-1}$ and the diffusion coefficient is as given by Bruce (1985):

$$D_B = c'e^{T d' + M f' + g'(T - h')^2} \quad (23)$$

with $c' = 0.32 \cdot 10^{-12}$, $d' = 4.35 \cdot 10^{-2}$, $f' = 17.1$, $g' = 1.8 \cdot 10^{-4}$ and $h' = 77.2$. Other known constants are $h_g = 2305 \cdot 10^3 \text{ J kg}^{-1}$ and $C_V = 3000 \text{ J kg}^{-1} \text{ K}^{-1}$. In one of Bruce's experiments (Bruce, 1985) with

barley, the air temperature was $T_a = 75^\circ \text{C}$, the initial moisture of barley was $M_0 = 0.271$ (dry basis), the initial temperature of barley was $T_0 = 33.3^\circ \text{C}$, and the air velocity was 0.56 m s^{-1} . The equilibrium moisture content, M_e , is taken to be 0.12 . D_0 is the value of D_B for $T = 30^\circ \text{C}$ and $M = 0.271$. The most suitable transfer coefficients are $h_T = 55 \text{ W m}^{-2} \text{ K}^{-1}$ and $h_m = 1.08 \cdot 10^{-6} \text{ m s}^{-1}$ according to model 2 of Miketinac et al. (1992).

In this paper, the diffusion coefficient is taken to be

$$D = D_B \left(\frac{\rho_s, t=0}{\rho_s} \right)^\gamma. \quad (24)$$

This is done to account for the fact that when the kernel shrinks, ρ_s increases but the diffusivity decreases. The positive parameter γ is not presently known from experiments and must be determined indirectly from the data by Bruce (1985) as outlined above by minimizing $J(\gamma)$ or $j(\gamma)$.

Since the expression for ρ_s found by Nelson (1986) is not a very accurate ($r^2 = 0.64$) representation of the data (which is not given in the paper by Nelson), the function S_V is used in a form given by Tang (1990) but β_V is determined from Nelson's expression for ρ_s in the following way. In the range of interest, i.e. $\bar{M} \in [0.1, 0.271]$, the function $S_V = e^{-\beta_V(M_0 - \bar{M})}$ is approximated by

$$S_V \approx 1 - \beta_V(M_0 - \bar{M}).$$

Then β_V is found by a least squares fit of this to the equation $S_V = \frac{\rho_{s,t=0}}{\rho_s}$ with ρ_s as given by Nelson. In this way it is obtained that $\beta_V = 0.36$. The value of β_V could have been determined simultaneously with the value of γ by using the inverse method. But the function $J(\beta_V, \gamma)$ (and also $j(\beta_V, \gamma)$) has several local minima and picking one of them as correct requires additional experimental evidence or theoretical reasoning.

Numerical experiments with both expressions (21) and (22) suggest that $\gamma = 20.00$ is the value for which these expressions are the smallest possible. When expression (21) is used, it is found that $J(0.36, 20.00) = 0.09999$ while in Miketinac et al. (1992) it was 0.097 (for model 2), and the maximum deviations are smaller than 4.5% for temperature and 4.2% for moisture while previously (i.e. model 2 in Miketinac et al., 1992) they were less than 4.5% and 3.5% respectively. This suggests that the fit of the experimental data is now slightly inferior to the fit given when the shrinkage was not taken into account.

From the data given above it follows that the exponent in the expression for the radius

$$\frac{R}{R_0} = S_V^{\frac{1}{3}} = e^{-\frac{1}{3}\beta_V(M_0 - \bar{M})}$$

is given by

$$\frac{1}{3} 0.36 (0.271 - 0.21) = 0.007,$$

which is small enough that $e^{-x} \approx 1 - x$. This implies shrinkage of 0.7% . For a column of barley 4 m high this amounts to shrinking

of 2.8 cm under the same drying conditions as above. Figure 1 shows that the shrinkage is small and linear with respect to time (at least during the first 300 seconds).

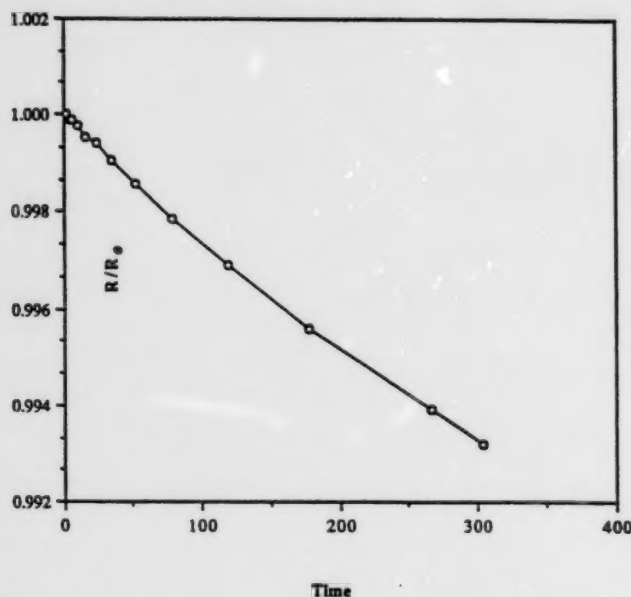


Figure 1. Ratio of computed radius and the initial radius versus drying time (in seconds).

Table 1 contains results of the analysis of the sensitivity of the computed solution, i.e. the surface temperature, T , and the average moisture content, $\bar{M}$, to changes in values of the two new parameters, β_V and γ . In this table all difference quotients are central differences; for example,

$$\frac{\Delta T}{\Delta \beta_V} = \frac{T(\beta_V + \Delta \beta_V) - T(\beta_V - \Delta \beta_V)}{2 \Delta \beta_V}$$

with $\Delta \beta_V \approx 0.2 \beta_V$, and so on. The sensitivities are evaluated after 300 s of drying. From the results in the Table 1, it is evident that both T and $\bar{M}$ depend more strongly on β_V than on γ .

Table 1.

parameter \ quantity	T	$\bar{M}$
β_V	1.736	0.035
γ	0.034	0.001

5. Summary and Conclusions

This paper contains a presentation of a model for the simultaneous heat and mass transfer in the drying of a shrinking kernel of barley. The presentation consists of three steps. The first step is a careful and detailed derivation of the model following the approach suggested by Aregba et al.. The second step is a numerical solution of the coupled system of partial differential equations modeling the drying process assuming that all parameters are known. The third step is to find optimal values of parameters that characterize shrinkage.

The model contains two parameters that characterize shrinkage since they did not appear in the previous models discussed by Miketinac et al. (1992). The cubical shrinkage coefficient is obtained from the experimental results by Nelson (1986) and the other parameter is obtained by minimizing the sum of squares of relative deviations of computed values from the experimentally observed values (Bruce, 1985) for the surface temperature and the average moisture content of barley.

Direct measurements of β_V (as in Tang, 1990) and the radius of the kernel (as in Misra and Young, 1980) are needed to further test the theory presented in this paper.

6. Nomenclature

h_{Jg}	heat of vaporization	$J kg^{-1}$
A	kernel surface area	m^2
C	specific heat	$J kg^{-1} K^{-1}$
D	moisture diffusivity	$m^2 s^{-1}$
h_T	heat transfer coefficient	$W m^{-2} K^{-1}$
h_m	mass transfer coefficient	ms^{-1}
K	thermal conductivity	$W m^{-1} K^{-1}$
M	local moisture content	decimal, dry basis
$\bar{M}$	average moisture content	decimal, dry basis
M_e	equilibrium moisture content	decimal, dry basis
r	radial coordinate	m
T	temperature	$^{\circ}C$
T_a	air temperature	$^{\circ}C$
t	time	s
V	volume	m^3
ρ	density	$kg m^{-3}$

Acknowledgements

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TECHNICAL SESSION 5

GEO-MECHANICS

Chair: J.F. Archibald



PHYSICAL AND NUMERICAL MODELLING OF OPENINGS IN CEMENTED HYDRAULIC BACKFILL

By

C. W. PELLEY

Department of Mining Engineering
and

R. J. MITCHELL

Department of Civil Engineering
Queen's University
Kingston, Ontario

ABSTRACT

A research project undertaken at Queen's University, evaluated the technical feasibility of utilizing openings in cemented hydraulic backfill as drill drifts for bulk mining stopes. As part of that project, physical and numerical modelling of such openings was carried out.

The physical modelling was carried out in the Queen's University 3m radius centrifuge on 60 and 120 scale models of openings in cemented base metal tailings. Homogeneous and layered models, some containing waste rock inclusions, were cast and loaded up to 180 gravities. This represents scale factors of 3 and 1.5 respectively for the models. Openings in homogeneous models were found to be stable and the waste inclusions caused no instability. Instability resulted from weak layers, especially when the models were undermined or exposed to closure.

Numerical modelling of a similar opening above an ore stope being mined in three lifts was carried out using the Fast Lagrangian Analysis of Continua code. This showed limited closure in the area of the opening and the opening to be stable until the sill beneath was removed. With removal of the sill, some backfill failed into the void and a new stable arch formed above the empty stope.

The results of both the physical and numerical models suggest that openings in homogeneous hydraulic backfill are stable without the passive support of a liner. The problems caused by the presence of weak layers can likely be lessened by the installation of an active support such as split-sets.

INTRODUCTION

Openings within backfill have been extensively used in cut and fill mining, both as extraction drifts for the overlying stope and as service drifts for the stope advancing from below. These openings in hydraulic fill have been established by installing support, usually timber support, and pouring fill over the supporting structure with drainage through a fabric liner. Support systems such as Bernold screen and Armco steel arches have been tried in certain locations. The most recent method in narrow ore utilizes cables anchored by split-sets to support the timber mat below the first pour of fill.

With the advent of bulk mining, using large diameter blastholes, the requirement exists for a drill horizon above the stope. If the ore above has been mined, this horizon needs to be protected from the open or filled stope above. The initial practice was to develop the drill horizon some distance below the stope above and to remove the sill with up holes at the end of the stope life. This process demanded additional development time and in some instances caused problems with stress build-up in the overlying pillar.

An alternative method would be to develop the drill horizon in the backfill of the overlying stope. Since secure access to the stope before filling is usually impossible, this drill access cannot practically

be maintained by placing support and filling around it. This means that the access must be excavated in the consolidated fill and adequately supported. This process has been tried successfully at Kidd Creek in cemented rock fill as described by Wittchen¹ and has also been experimented with in hydraulic fill at Inco's Creighton Mine using an Alpine miner to excavate the backfill and timbersets for support.

A research project was carried out at Queen's University, funded by the Mining Research Directorate, to evaluate the practicality of stope re-development through cemented hydraulic fill. Since the support requirements of such a drift are significant in terms of the cost and excavation time, physical and numerical modelling was carried out to assess the stability of such openings.

THE PHYSICAL MODELS

One problem associated with creating openings in hydraulic backfill is the installation of support in the rounded tunnel outline. Two possible situations necessitating support are the existence of inclusions of waste material which may be present above the tunnel opening and the existence of weak layers of fill caused by cycles in the backfill pour. To better assess the stability of such openings and any instability caused by inclusions or weakness planes, centrifuge modelling of backfill openings was carried out.

Scale models approximating the actual situation were created with gravitational load applied beyond the scale factor by spinning the model in a centrifuge. The centrifuge used for this modelling was a 3 meter radius, 30 g-tonne machine built at Queen's University in 1985¹. The situation modelled was a 5.2 meter wide by 4.5 meter high opening in cemented backfill above a 15 meter stope as shown in Figure 1. The models created were 60 and 120 scale models using a truncated circular opening of 50 and 25 mm radius as shown in Figure 2. This equates to a 6 m wide tunnel at both scale factors. The fill width scales to approximately 18 meters in the 60 scale model and to 36 meters for the 120 scale models. All models which were tested are described in Table 1.

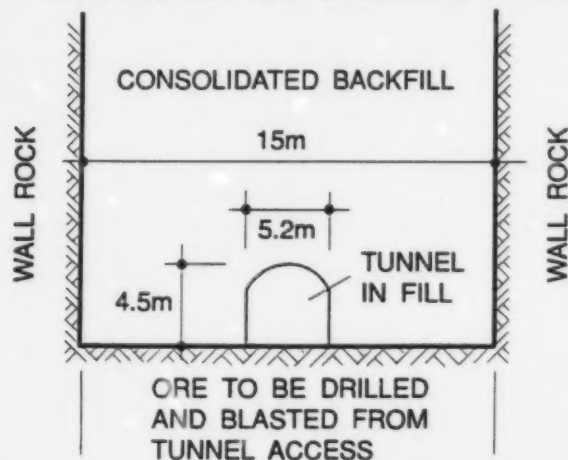


FIGURE 1: PROPOSED PROTOTYPE DEVELOPMENT.

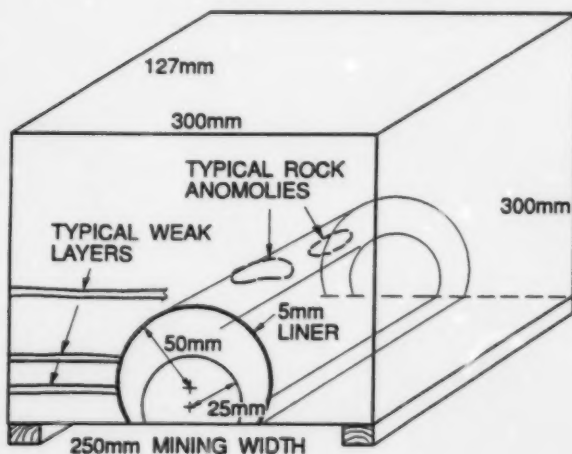


FIGURE 2: NOMINAL MODEL DIMENSIONS.

MODEL MATERIAL

The tailings material used in this modelling study is a cycloned backfill product supplied by Falconbridge Ltd. from its Strathcona mill operation. Prior to testing two samples were collected for sieve analysis and the results are shown in Figure 3. The results are quite standard for base metals backfill material.

MODEL PREPARATION

Boxes in which to pour and cure all models were constructed from treated formply. Lucite tubing was cut and formed to create molds for casting the concrete liners for models 30-2, 30-4, 30-6, and 30-7. These liners were precast by vibrating a thickened sand-cement grout slurry into the molds. The compressive strength of control samples formed of the liner mix was found to be about 10 MPa.

TABLE 1: MODEL DESCRIPTION AND RESULTS SUMMARY.

MODEL NO	TEST	FAILURE MODE	SIMULATED GRAVITIES AT END OF TEST	CONTROL UCS KPa
60-1	Tunnel Stability (homogeneous)	None	180	192
60-2	Tunnel Stability (anomalies)	None	180	166
60-3	Tunnel Stability (weak layers)	Caving of Top of Tunnel	108-145	172
60-4	Undermined (homogeneous)	Caving to Top of Tunnel	15-22	137
60-5	Undermined (homogeneous)	Collapsed on Weak Layer	11	111
60-6	Undermined (homogeneous)	Caving Extending Above Tunnel	11-51	126
30-1	Tunnel Stability (homogeneous)	None	180	376
30-2	Tunnel Stability (homo., lined)	None	180	347
30-3	Tunnel Stability (anomalies)	None	180	350
30-4	Undermined (homo., lined)	Caving to Mid-tunnel	68	360
30-5	Tunnel Stability (weak layers)	Caving to Weak Layers	120-190	397
30-6	Closure Stress (layered lined)	Initial Crack at 0.7% Strain Badly Cracked at 2% Strain		418
30-7	Closure, Undermined (homo., lined)	Caving to Mid-tunnel	70	380

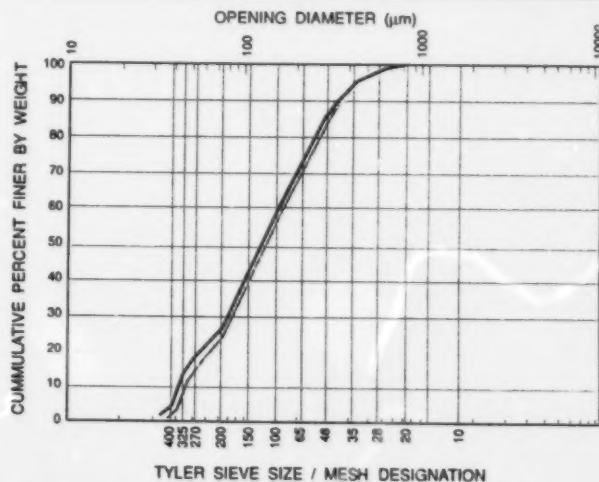


FIGURE 3: SIZE DISTRIBUTION OF TAILINGS PRODUCT

All centrifuge models were formed from 16:1 tailings:cement ratio (60 scale models) or 25:1 ratio (120 scale models) mixes poured at a pulp density of 70%. Anomalies and weak layers, as noted on Table 1 and Figure 2, were incorporated in selected models. Normal Portland cement was used to pour the models and these were cured for 28 days in a warm misted curing facility. All weak layers were poured at a ratio of 40:1 with average UCS of 110 kPa. Stone anomalies were 65 mm and 30 mm maximum dimension, for the 60 and 120 scale models, respectively, and had a specific gravity of 2.7.

MODEL TESTING

Models were placed, after 28 days of curing, into a centrifuge strongbox which included a hanging wall plate for application of a closure stress or displacement. Figure 4 shows a model in the strongbox. The normal test procedure was to increase the centrifuge speed in increments while visually monitoring the stability performance of the model through a transparent cover plate on the strongbox. Visual monitoring was facilitated by a digistrobe and video camera

arrangement. Visible surfaces of the models were spray painted to improve the image and allow immediate identification of areas where surface spalling developed.

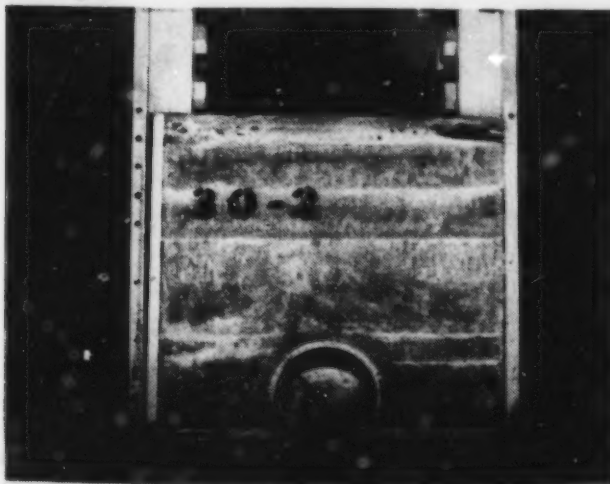


FIGURE 4: MODEL WITH LINER IN STRONGBOX.

The purpose of centrifuge model acceleration is, of course, to simulate prototype gravitational (self-weight) stresses by the induced inertial stresses created by centripetal force on the model. Various results of mine backfill stability modelling have been reported by Mitchell^{3,4}. The Queen's 30 g-tonne geotechnical modelling centrifuge creates 60 gravities on a model at a centrifuge speed of 141 rpm and 120 gravities at a centrifuge speed of 200 rpm. Models were accelerated at increasing speeds either until a failure was visually observed or until the speed was 250 rpm. At 250 rpm the induced simulated gravitational acceleration is over 180 gravities. This is three times the model scale for the smaller tunnels and 1.5 times the model scale for the larger tunnels, indicating that models which did not fail had static safety factors of at least 1.5 or 3 against failure. Where failures were observed at a simulated gravitational factor of N gravities it is indicative of prototype failures in cases where the prototype dimensions are equal to N times the model dimensions. The model testing results are presented in Table 1.

TUNNEL STABILITY MODELS

Models 60-1 and 60-2 showed no failure and demonstrate that tunnels in the homogeneous 25:1 backfills, having unconfined strength (UCS) of 166 kPa would be stable at the prototype dimensions with a safety factor of at least 1.5. Model 60-3 was the first of this series of models to show distress, exhibiting some superficial spalling at a scale factor of about 110 (close to the prototype scale of 120) and a minor roof collapse at a scale factor of 145 (220 rpm). The distress appeared to be controlled by and limited to the weak 40:1 layers created in this model. This model demonstrates that weak layers could pose a potential minor caving risk at the prototype scale envisaged for stope development in a weakly cemented backfill.

The large tunnel models (60 scale) were poured using 16:1 backfill having an unconfined strength in the order of 350 kPa. Model 30-1 confirmed that prototype scale tunnels in a backfill of this strength would be quite stable. Slight tunnel side spalling was noted at a scale factor of 180, which is equivalent to having a prototype tunnel of three times the design dimensions. It can be assumed that the selected tunnel would have a safety factor of at least 3 against distress. Model 30-2 was tested to well over 180 gravities to show that the model liner was compatible with fill self-weight deflections. No cracking or other distress signs could be observed in this model liner. The model liner represents, approximately, a 20 MPa prototype concrete liner of 15 cm thickness.

Model 30-3 confirmed the findings from model 60-2, that buried rock anomalies would have little or no effect on tunnel stability. Model 30-3 was made larger, after initial testing, to expose the bottom side of one such anomaly and was tested to well over 180 gravities. This exposed anomaly model is shown in Figure 5. These anomalies represent, at prototype scale, 2 m by 4 m chunks of wall rock buried close to the top of the tunnel. This strongly suggest that such anomalies would not affect the stability of a tunnel in an otherwise homogeneous backfill although they would create tunnel excavation problems.



FIGURE 5: EXPOSED ANOMOLY IN MODEL 30-3.

Model 30-5 is shown, after testing, in Figure 6 and again demonstrated that weak layers can be a cause for concern. In this case, however, simulated gravity (included) stresses of 2 and 3 times the expected prototype mass gravity stresses would be required to cause spalling or caving of weak layers in this generally stronger 16:1 fill. All weak layers were poured at 40:1 but it is expected that stronger adjacent backfill will lend greater support to weak layers in the fill. Thus, these model studies indicated that the proposed prototype stope development tunnels could be safely created in consolidated backfills having unconfined strengths (UCS) greater than about 250 to 300 kPa, even if rock anomalies and some limited extent of weak layering were to be found in the fill.



FIGURE 6: FILL STABILITY DURING SILL REMOVAL.

FILL STABILITY DURING SILL REMOVAL

Model 60-4 experienced failure by a caving collapse at a scale factor of 30 (100 rpm) with a 25 cm clear unsupported span. This is equivalent to undermining a sill of 7.5 m width. Model 60-5 indicates that caving can develop at even lower spans if the fill contains weak layers. Model 60-6 experienced failure by a major caving collapse at a scale factor of 50 (130 rpm) with a 17 cm clear span. This failure, shown in Figure 7, occurred at a prototype span of 8.5 m. Thus, it is concluded that the 25:1 mix is not sufficiently strong to resist dilution caving when undermined to spans exceeding about 7 m.



FIGURE 7: CAVING OF UNDERMINED FILL

Model 30-4, experienced a caving collapse at a scale factor of 68 which is equivalent to a prototype clear span of 17 m. Model 30-7 was subjected to a closure strain of 1% (i.e. closure of 3 mm) and exhibited caving collapse at a scale factor of 75. Thus, it appears that closure stress, within the elastic limit, has little effect on the fill caving phenomenon. It can be concluded, from these models, that the 16:1 backfill is marginally stable and is unlikely to cause any serious dilution by gravity failure if undermined to the prototype span of 15 m. The ratio of UCS/stable span is found to be about 15 for the 120 scale models and 21 for the 30 scale models. This may be indicative of some second order scaling effect but is more likely to be related to the fact that the ratio of tensile to compressive strength is generally found to be higher for better quality backfills.

Model 30-6 was used to examine the effect of layering on the stability of backfills under closure strains. Cracking along a weak layer was noted at closure strains much less than 1% and that major yielding of the fill occurred between 1% and 2% closure strain. There is little doubt that high closure strains would result in significantly increased dilution during sill removal, especially if the backfill contains weak layers.

NUMERICAL MODELLING

To complement the physical modelling carried out in the centrifuge, a numerical model was created using the FLAC code as available from Itasca, of an opening in backfill and an extraction sequence based on using that opening as a drill drift for a bulk mining stope. It was hoped to find agreement with the physical models and to evaluate the degree of closure which would occur before the final stope blast occurred.

The model used is a 120 meter high by 60 meter thick cross-sectional model with a vertically dipping 10 meter thick ore zone running through the middle. The input parameters are shown in Table 2 and the model depicted in Figure 8. The backfill properties are for a 20:1 tailings to cement product cured for 112 days. The rock parameters were modelled as elastic and the backfill as Mohr-Coulomb. The excavation sequence is shown in Table 3.

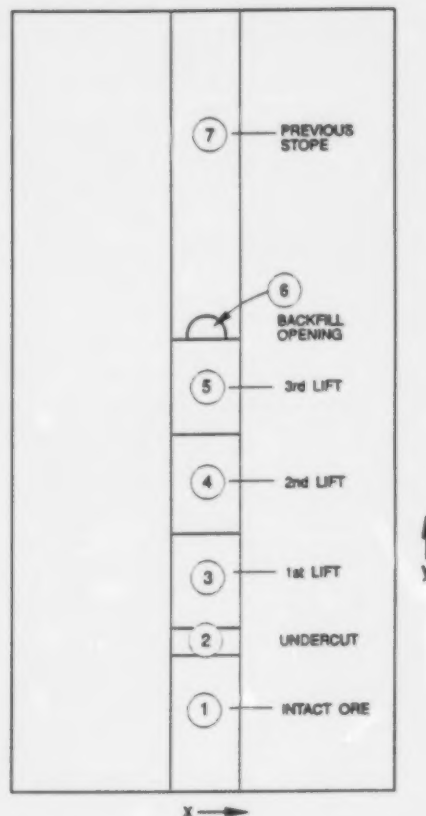


FIGURE 8: SECTIONAL FLAC MODEL OF OPENING IN BACKFILL

TABLE 2: MODEL INPUT PARAMETERS

Depth of Burial	1000 meters		
Stress Gradient			
Vertical	0.27 MPa/meter		
Horizontal	1.8 x vertical		
Elastic Parameters	Young's Modulus	Poisson's Ratio	Density
Ore	40,000 MPa	0.18	3.2
Waste	34,000 MPa	0.20	2.7
Backfill	250 MPa	0.35	2.0
For Backfill	Friction Angle = 35° Cohesion = 175 kPa Tensile Strength = 70 kPa		

TABLE 3: MODELLING SEQUENCE

STEP	EXCAVATION PROCESS	# OF ITERATIONS
1	Excavate the Upper Stope (7)	1000
2	Backfill the Upper Stope (7)	1000
3	Excavate Backfill Opening (6) and the Undercut (2)	1000
4	Excavate 15 meters of Stope (3)	2000
5	Excavate 15 meters of Stope (4)	2000
6	Excavate Remaining Ore (5)	Failed at Step 6340

ANALYSIS OF NUMERICAL MODELLING RESULTS

One of the most convenient aspects of the FLAC numerical analysis code is the ability to keep a history of up to 25 points during the modelling process. In

addition coloured contour plots are available at the end of each step if the results are saved. This analysis is based on these graphics as provided by FLAC output.

The main results of interest are the stresses in the rock below the backfill opening, the closure which occurs at or above the opening, the stresses in the backfill above the opening, and any vertical displacement which occurs above the opening. It should also be noted that the absolute magnitude of these values are considered high as the model assumes no transfer of stress to abutments on the third dimension. Of more importance are the relative values during the excavation process.

The closure in the initial stope, 20 meters above the pillar elevation is shown in Figure 9. It shows rapid closure during the first 2000 time steps, that is, during the first 1000 as an open excavation and 1000 after the backfill is placed. Closure is almost nil during the remaining period until the pillar is excavated at time step 7000. The total closure on this wall is 250 mm. Since the same would occur on the opposite wall this represents approximately 5% closure. This is reflected in the horizontal stress as shown in Figure 10, at a point 8 meters above the backfill opening. Stress levels have increased to 11 MPa by the step 2000 when the backfill opening is created. It increases only slowly to peak at just over 14 MPa before dropping rapidly as the backfill failure into the open stope occurs. These values are considered high considering only 5% closure has occurred.

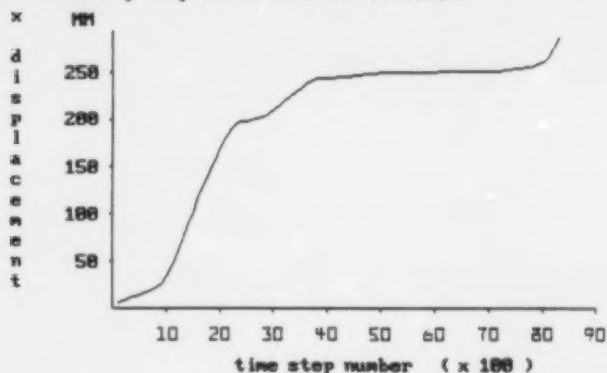


FIGURE 9: CLOSURE 20 METERS ABOVE BACKFILL OPENING

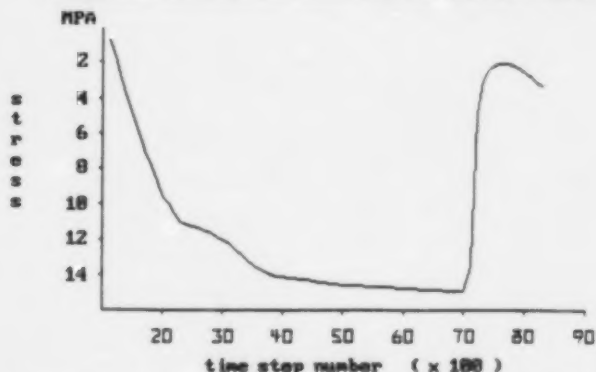


FIGURE 10: HORIZONTAL STRESS IN BACKFILL ABOVE OPENING

Figure 11 shows the vertical stress at the same point. It has increased to just over 6 MPa at step 2000, drops slightly as some relaxation occurs into the backfill opening, then stabilises again until failure occurs.

Closure adjacent to the backfill opening as shown in Figure 12 is very small, some 20 mm maximum from steps 2000 to 7000. Likewise the vertical displacement at a point 2 meters above the opening is very low as

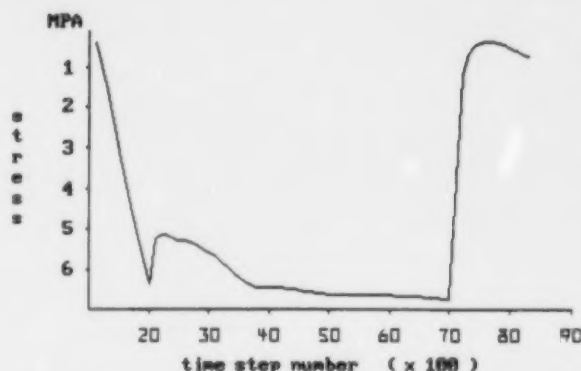


FIGURE 11: VERTICAL STRESS IN BACKFILL ABOVE OPENING

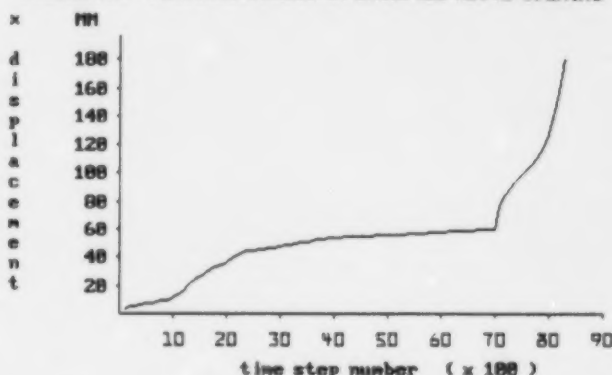


FIGURE 12: CLOSURE ADJACENT TO BACKFILL OPENING

shown in Figure 13. There is some displacement as the opening is excavated then little movement until failure occurs.

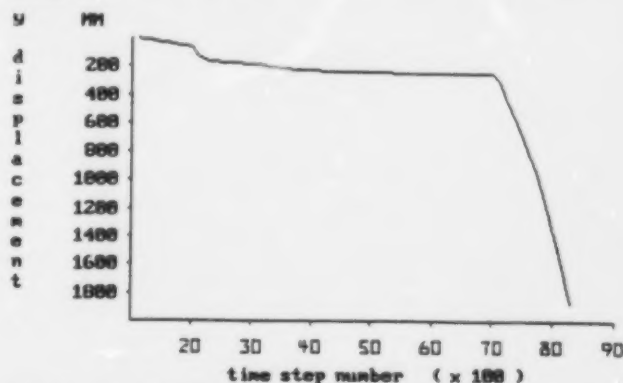


FIGURE 13: VERTICAL DISPLACEMENT 2 METERS ABOVE OPENING

The horizontal stress distribution in the pillar abutment adjacent to the backfill opening is shown in Figure 14. It shows an approximate 20 MPa increase in stress during the mining of the bottom two-thirds of the stope. The stress mid-point in the sill pillar is shown in Figure 15 which builds to a maximum of approximately 160 MPa. Stress contours of the area before removal of the sill are shown in Figure 16. The edge effects peak at 450 MPa but generally the core of the pillar is approximately 150-200 MPa. Again these are considered high as some shedding would occur to pillars along strike.

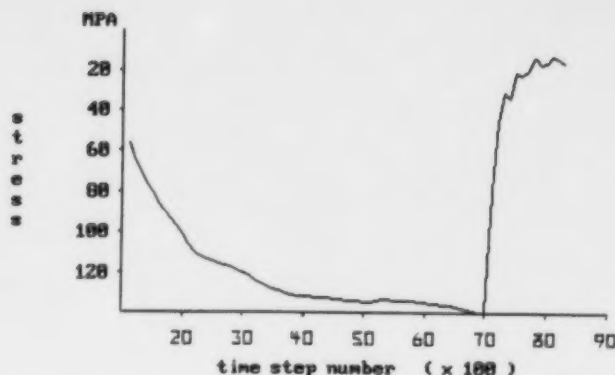


FIGURE 14: HORIZONTAL ROCK STRESS ADJACENT TO OPENING

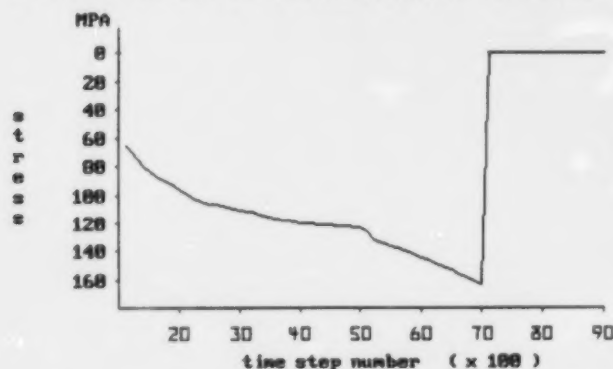


FIGURE 15: HORIZONTAL STRESS IN PILLAR BELOW OPENING

STRESS CONTOURS

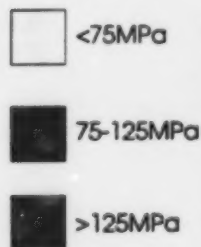


FIGURE 16: HORIZONTAL STRESS DISTRIBUTION BELOW OPENING

Finally, Figure 17 shows the fill failure which occurs into the open stope after the sill pillar has been removed. The interesting point here is the re-establishment of a stable arch as shown by the $<5\text{mm}$ of movement contour. In fact this image is remarkably similar to Figure 7 showing the re-establishment of a stable arch in the centrifuge model after the support had been removed.

SUMMARY AND CONCLUSIONS

Both physical and numerical models have shown that a circular opening having a diameter of 5-6 meters in homogeneous cemented hydraulic backfill can be considered to be stable. The cement to tailings ratio of the backfill used in the physical models ranged from 16:1 to 25:1, and the properties used in the numerical model equated to a 20:1 fill material.

DISPLACEMENT CONTOURS

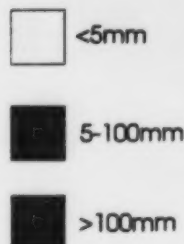


FIGURE 17: BACKFILL FAILURE AND FORMATION OF A STABLE ARCH ABOVE THE OPENING

The physical models have shown that rock inclusions buried in fill above the opening do not seriously endanger the stability of the opening. Inhomogeneous layering, which commonly exists in normal backfill pours, does affect the stability. It is suggested that as the overall strength of the backfill is increased, this effect is lessened. The stability could be improved if the backfill in the area of a potential opening is poured in one cycle or if some support mechanism such as split-sets were used to hold the layers together.

Closure strains applied to the physical models showed rapid deterioration of the fill stability. The numerical models showed little closure in the backfill immediately above the rock pillar but rapid failure as closure took place after pillar removal. Both physical and numerical models showed re-establishment of a stable arch above the undermined area. The failed material would cause dilution but this is similar in any mining situation where backfill is undermined. The dilution would be reduced by using a higher strength material.

A high level agreement exists between the physical and numerical models and shows these methods to be valuable in predicting the stability of openings in a homogeneous backfill material.

ACKNOWLEDGEMENTS

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The Use of Thin Spray-on Linings as an Alternative Technique for Conventional Underground Support

by

James F. Archibald, Associate Professor
Robert A. Mercer, Graduate Student
Peter Lausch, Research Assistant

Department of Mining Engineering
Queen's University at Kingston

ABSTRACT

An innovative form of underground support has been subjected to review by staff of the Department of Mining Engineering, Queen's University at Kingston, under funding provided by the Mining Industry Research Organization of Canada (MIROC). This support technique utilizes spray application of a solid polyurethane membrane, designated Mineguard™ by its manufacturer, onto rockwalls and backs within underground excavations. The membrane material is planned as a replacement for shotcrete and bolt and screen support techniques which are commonly used within mines in order to provide secondary wall support. Where sufficient layer thicknesses and complete sealing of rock surfaces may be effected, it is anticipated that additional support resistance against gravity-induced back failure may also be developed through provision of the "Caisson" support effect. Both laboratory and in-situ performance reviews of Mineguard™ have demonstrated that appreciable support capability against gravity-induced failures in walls and backs, exist in terms of load deformation response, failure strength and surface adhesion behaviour.

1.0 INTRODUCTION

This paper documents progress in the development of thin, spray-on polyurethane liner materials for use as an alternate method of ground control. Research conducted thus far has been coordinated by staff of the Queen's University Mining Engineering Department and has involved several phases, including extensive laboratory testing and underground testing.

The original project goal was to develop a form of support that, by utilizing the Caisson Effect, could effectively compete with thin shotcrete lining and bolts and screen support techniques. Development of the Caisson Effect requires creation of air impermeable coatings about underground excavations. After coating, any subsequent deformation of the rockmass increases the air volume within rockmass voids, thereby decreasing the void pressure and creating a pressure imbalance across the coating surface. The resulting pressure imbalance has been shown, solely on the basis of laboratory trials, to be able to support significant rock loads^(1,2), and has been used to help explain the effectiveness of thin shotcrete liners⁽³⁾.

Several candidate materials were originally identified and were thought to be capable of supplying the air impermeability conditions necessary to promote the Caisson Effect. Of the candidates, the most promising proved to be a two part polyurethane using high pressure plural component equipment for spray in place application. The reaction and subsequent "set" occurs in seconds. This two-part polyurethane material has been

designated Mineguard™ by its manufacturer, Urylon Plastics Inc. of Guelph, Ontario.

It was recognized during the initial research phases that application of plastic materials underground could present a potential fire hazard. To alleviate this problem, three variations of the original Mineguard™ formulation were developed and were given the designations Mineguard™2 through 4. Mineguard™2 comprises the original formulation with the addition of a chemical fire retardant; Mineguard™3 consists of Mineguard™2 with the addition of a fire-resistant vermiculite coating; and Mineguard™4 consists of Mineguard™3 to which a second, emulsion-based vermiculite layer is installed. Each Mineguard™ variation was classified in terms of flame spread rating, or flammability potential, at the Underwriters Laboratories facilities in Scarborough, Ontario using National Bureau of Standards Construction Guidelines. On the basis of such tests, Mineguard™3 exhibited the best flame spread rating, at a level well below that recommended for residential construction materials and approaching that of gypsum wall board, which is normally considered to be non-flammable⁽⁴⁾.

Further tests were conducted to determine the basic engineering properties of the Mineguard™ material, including blast durability evaluations conducted at the Queen's University explosive test site facility. The results of this particular test suggested that the material should have sufficient durability to sustain similar punishment underground. With encouraging results demonstrated by initial flame spread and surface blast durability tests, physical

response and applicability trials were conducted within mine environments.

2.0 UNDERGROUND TESTING

2.1 Mineguard™ Installation Equipment:

Equipment used for Mineguard™ installation in all underground tests consisted of off-the-shelf equipment that was not modified in any way for underground use. In terms of size, the most significant system components consisted of two 200 litre drums, each containing one of two liquid chemical agents that, when mixed, react to form solid polyurethane. The liquid components, referred to as side A and side B, consist of an isocyanate and polyol/ resin chemical, respectively. The side B agent also contains whatever additives are required to alter the polyurethane physical properties. Such additives may include catalysts necessary to speed up the reaction time between liquid and solid phases and/ or chemicals to improve material fire retardancy. For spray installation, the chemicals must be preheated to 50° C, then pumped under pressure at a fixed volumetric ratio of 1 : 1 through separate hoses to the spray gun. When sprayed onto a surface, the heated liquids form a thin but continuous lining which sets to a high proportion of its ultimate strength within approximately 5 seconds. The spray gun, weighing less than 5 kg, can be attached to the pumps by up to 50 metres of hose. This provides the applicator with a high degree of mobility, although a scissor truck would be required to permit spray access to the excavation back in order to maintain a 1 metre spray distance between the gun and the surface to be coated. The 1 metre offset distance is that which is normally recommended by commercial applicators of polyurethane materials.

The applicator is required to wear safety equipment that includes disposable coveralls and breathing apparatus. The breathing apparatus, as in all cases where chemical spraying is conducted, is required to protect the applicator from possible contact with local concentrations of airborne contaminants. Whereas typical spray operations require only single operators, the need to spray vermiculite surface coating material (for Mineguard™3) presently requires that a second operator follow behind the applicator, blowing vermiculite onto Mineguard™ coated surfaces prior to its complete cure into the solid state.

Table 1: Summary of the Underground Testing

	TEST SITE LOCATION			
	Kidd Creek		Inco 1	Inco 2
	Stage 1	Stage 2		
Site Depth Below surface (ft)	2800		2600	2600
Excavation Site	Air-receiver chamber		2105 Crosscut	2280 Crosscut
Material Used	Mineguard™2		Mineguard™3	
Area Covered (m²)	215.5	872.5	115.0	93.1
Average Rate of Application (m²/ min.)*	1.23	1.63	1.47	1.77
Average Layer Thickness (mm)	2.87	1.66	1.58	1.68
Application Cost (\$/ m²)	31.01	17.87	17.04	18.21

* intermittent spray application, "stop and go" activity

2.2 Underground Testing Program:

The objectives of the underground testing phase of this project can be summarized as follows:

- to evaluate the suitability of existing polyurethane application procedures for underground use,
- to monitor air-quality and contamination potential offered by chemical agents during and after the Mineguard™ installation process,
- to accurately assess application conditions, such as rates of installation, site preparation needs and degree of coverage,
- to assess material installation cost factors, and
- to evaluate the product's long and short term durability.

The three underground spray tests completed to date were undertaken between December, 1990 and June, 1991. The first test was conducted at the Kidd Creek Mine in Timmins, Ontario and involved spraying within a large air-receiver chamber (in two stages) over two weekends. The air-receiver chamber is designed to act as a reservoir for compressed air. In order to effectively maintain pressure in fractured ground, it was required to completely seal the excavation with an air impermeable lining. Thick (2.5 - 3.0 mm) layers of Mineguard™2 were used to provide this lining. While spraying was being conducted, extensive air-quality measurements were undertaken by the Ontario Ministry of Labour. On the basis of such measurements, no air quality hazards were identified to exist relative to Ontario Ministry of Labour guidelines⁽³⁾.

Subsequently, two additional underground tests were undertaken at Inco's Copper Cliff North Mine in Sudbury, Ontario. Both trial sites were located in crosscuts that were to be used as draw points during mine production. Because possible risks of fire could be anticipated, as a result of ordinary mining activity in the crosscuts, Mineguard™3, due to its low flame spread character, was applied in these sites.

The results of the underground testing are summarized in Table 1.

Based upon data listed in Table 1 it can be seen that material cost is directly proportional to the thickness applied. Tests also showed that Mineguard™ application rates are high and improved from trial to trial as workers became more proficient at spraying underground. It should be noted that the stated application rates are representative of "stop and go" activity and, while spraying continuously, application rates were found to be significantly higher than those which are reported in Table 1.

As earlier indicated, one of the objectives of underground testing was to evaluate the durability of the product. Because Mineguard™ had demonstrated its ability to withstand severe blast damage in preliminary surface tests, it was decided that full scale blast durability tests would also be conducted in-situ during both Inco trials. These tests involved blasting a normal 4 metre development round directly onto Mineguard™ coated crosscut walls and backs.

In the first blast trial, the thin polyurethane lining was observed to be extensively damaged by ejected rock. Multiple penetrations and some large tears of the coating, particularly within a two metre offset distance from the blast source, were observed. Upon further investigation, it was suggested that the material's poor blast durability was due, in part, to excessive ventilation being blown into the crosscut during spray application. The ventilation rate produced large amounts of overspray loss and possible component imbalances within the polyurethane material as it was sprayed onto the excavation surfaces.

In the second Inco site trial, ventilation was maintained at normal mine flow rates and the overspray problem was alleviated. However, following blast initiation, the Mineguard™ lining was again shown to sustain impact damage similar to that which had been observed in the first Inco trial. Upon closer inspection of torn Mineguard™ layers, it was assessed that the lining exhibited less than designed physical strength characteristics. Microscope examination of retrieved in-situ samples showed the presence of a "foam-like" cellular structure within the lining which would clearly affect both the material and adhesion strength properties of the lining. Such "foaming" of the polyurethane lining material could have been due to the presence of excessive moisture in the air, which would react with the isocyanate chemical component to form CO₂.

The CO₂ released could then have become entrained as bubbles within the reacted, solid polyurethane layer, possibly forming the observed cellular structure. The comparatively humid conditions experienced underground during both Inco trials therefore may have resulted in the observed deterioration of the Mineguard™ material which resulted in the lining being unable to resist blast impact effects. Similar deterioration of Mineguard™ materials was not observed to occur within coatings installed during the Kidd Creek trial, where lower relative humidity conditions were thought to exist.

3.0 LAB TESTING

The results of three underground spray trials of Mineguard™ polyurethane material, in general, and the poor results observed during the underground blast durability tests, in particular, resulted in the initiation of two further phases of laboratory review. These two investigations were designed to answer specific concerns including:

a) what is the effect on material and adhesion strength of elevated humidity levels encountered during spraying;

b) what changes can be made to the material (if necessary) in order to mitigate the effects of spraying within sites known to be affected by elevated humidity levels;

c) what is the effect on adhesion strength of spraying Mineguard™ onto different rock substrate surfaces;

d) what load resistance can be generated by the liner in response to rockmass deformation; and

e) what is the dominant mechanism by which load resistance is generated.

A series of pull tests were therefore conducted in order to address the first three concerns while a series of block deformation tests were designed to address the last two questions. A description of each test will be provided in the following sections.

3.1 Pull Test Methodology

A pull test apparatus, depicted in Figure 1, was designed and manufactured to measure load resistance generated by thin polyurethane layers in response to being pulled perpendicular to the surface upon which they were sprayed. For laboratory tests, substrates which were coated consisted of 7.5 cm thick concrete slabs.

Pull tests were conducted using two types of 0.3 cm thick by 25 cm diameter steel pull-plates. One plate type was solid and included no holes in its surface except for holes necessary to mount pull screws. For installation each solid plate was initially placed flush against the concrete substrate. Both it and the substrate were subsequently sprayed over. In this test, no polyurethane material was permitted to exist between the plate and the substrate surface. Therefore, when the Mineguard™-coated plate was pulled directly away from the substrate surface, a "measure" of material strength was obtained.

The second type of plate that was utilized was a perforated pull plate. This plate was identical in size to the solid plate form except that its surface was perforated with holes to permit sprayed polyurethane (Mineguard™) to flow between the plate and the substrate, therefore permitting the material to bond effectively to substrate material beneath the plate. The application procedure in this case involved spraying an initial thin layer of Mineguard™ over the substrate surface. Before this initial layer was able to cure (<5 seconds) the perforated plate was pushed into the liquid

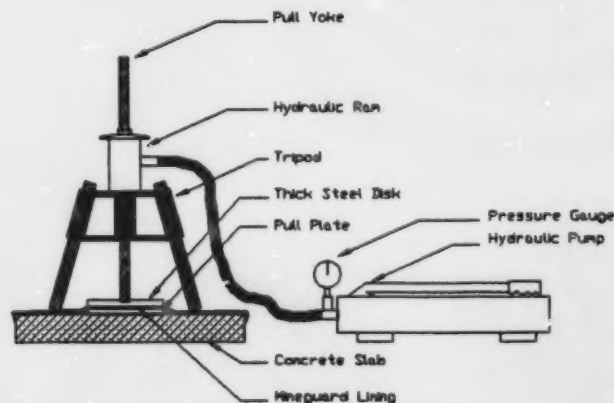


Figure 1: Pull Test Assembly

material which subsequently "set". The plate and substrate were then coated over by applying a second liquid material layer. In this case, when the plate was pulled away from the substrate, the bond that had been formed beneath the pull plate between the initial polyurethane lining and the substrate surface failed, giving a measure of liner adhesion to the substrate.

The basic components of the pull test apparatus (Figure 1) consisted of a hydraulic pump, a tripod-mounted hydraulic ram, a threaded pulling yoke and a heavy steel disk. The heavy steel disk was attached via screws to the previously-described pull plates to uniformly distribute pull forces over the entire surface area of the plate assembly. As the hydraulic ram was expanded, the plates were pulled away from the substrate to provide a measure of resisting load (measured on a calibrated dial gauge) versus liner deformation (measured by a hand-held scale).

Typical load versus deformation curves for each of the two plate types used are illustrated in Figure 2. It can be seen from these curves that much higher peak loads were generated initially by the perforated pull plates due to significant adhesion strength developed beneath each plate. However, once adhesion bond failure was shown to be achieved, very similar load resistance was generated by both types of plates due to the combination of adhesion and material strength of the thin liners existing around the perimeter of the pull plates.

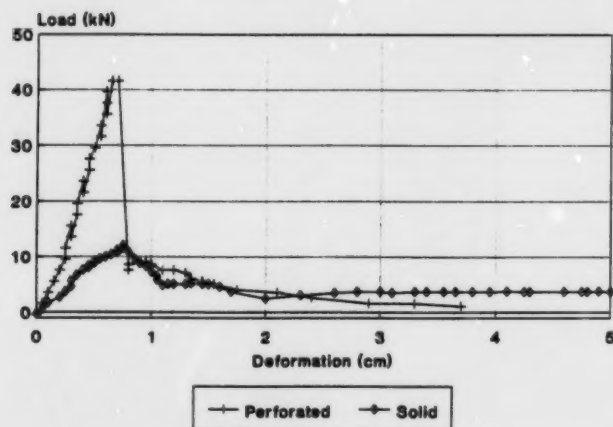


Figure 2: Typical Load-Deformation Response Curves for Pull Plate Tests

3.1.1 Effect of Humidity on Lining Strength Behaviour

The pull test apparatus was used to determine the effects of humidity on material and adhesion strength properties. The substrate material chosen for all humidity tests consisted of 45cm x 45cm x 7.5cm thick concrete slabs. Such concrete slabs were selected as the substrate material of choice to ensure that any differences in load resistance characteristics of the spray-on lining could only be attributed to variations in the linings properties and not to differences in substrate surface conditions, strength and/or roughness.

All spray tests were conducted within a humidity-controlled spray booth owned by Urylon™ Plastics Inc. of Guelph, Ontario. Both temperature and relative humidity levels were continuously monitored during all stages of spraying using a Vaisala Humicap humidity meter calibrated to accuracy levels of $\pm 1^\circ\text{C}$ and $\pm 1\%$ R.H., respectively.

The effect of humidity on material and adhesion strength has

been summarized and illustrated in terms of stress vs. humidity graphs. In perforated pull plate tests, adhesion strength was calculated by dividing the measured peak loading force applied to coated pull plates by the area of adhesion failure measured to exist beneath each plate. In the case of solid pull plate tests, the material strength was calculated by dividing the peak loading force applied in each test by the product of the plate perimeter and measured thickness of the lining. As can be seen from the summary graphs included in Figures 3 and 4, average Mineguard™ adhesion strengths were observed to decrease significantly at relative humidity levels greater than approximately 66 %. Average Mineguard™ material strengths, determined by

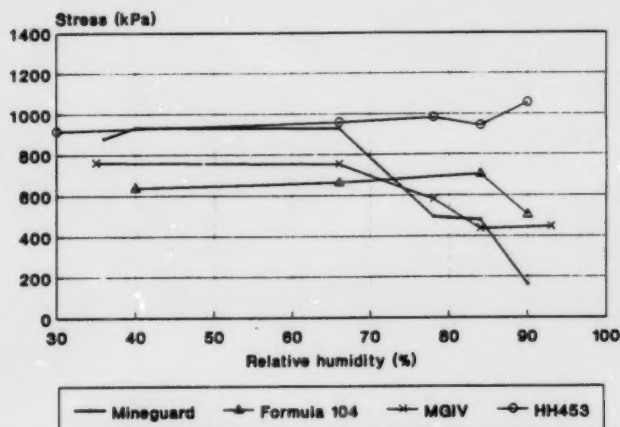


Figure 3: Adhesion Strength vs Humidity

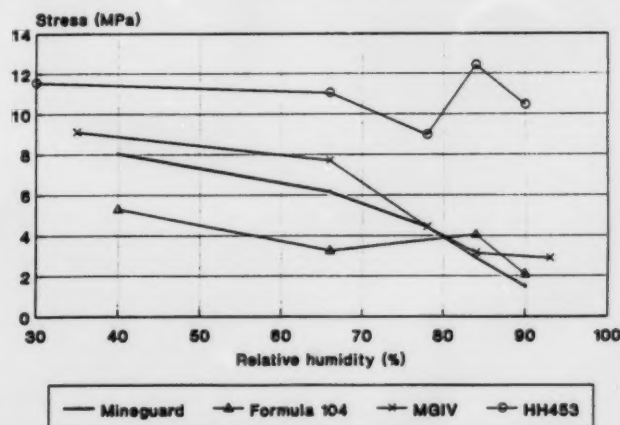


Figure 4: Material Strength vs Humidity

solid pull plate tests, appeared to decrease uniformly with increased humidity at all levels above site ambient conditions, this being in the range of 30-40% R.H. The apparent deterioration of basic material and adhesion strength properties at elevated humidity levels was found to be consistent with observations made during underground installation tests. Therefore, the poorer than expected blast resistance capabilities of the Mineguard™, observed to occur at both Inco in-situ installation trials could have, at least in part, been due to this apparent decrease in material and adhesion strength properties with increased humidity conditions

3.1.2 Review of Humidity Mitigating Techniques:

Since it can be anticipated that many potential underground application sites could experience elevated humidity levels that would fall within the range where Mineguard™ material properties have been shown to be deteriorate, it was proposed to investigate methods for chemically mitigating the effects of elevated humidity.

Urylon™ Plastics Inc. developed three chemical variations of the original Mineguard™2 formulation in attempts to make a humidity resistant Mineguard™ formulation. Each of the proposed humidity-resistant formulations was subjected to laboratory pull test evaluation in a similar fashion to tests conducted upon the original Mineguard™ formulation, and under varying relative humidity conditions.

The results of pull tests upon the three alternate formulations are presented and compared versus the results of the original Mineguard™2 tests in Figures 3 and 4. In addition Figure's 5 a), b) and c) provide a comparison of tensile strength, Young's Modulus and % elongation behaviour of the four Mineguard™ variants versus humidity. Several observations can be made from these figures the most notable of which are:

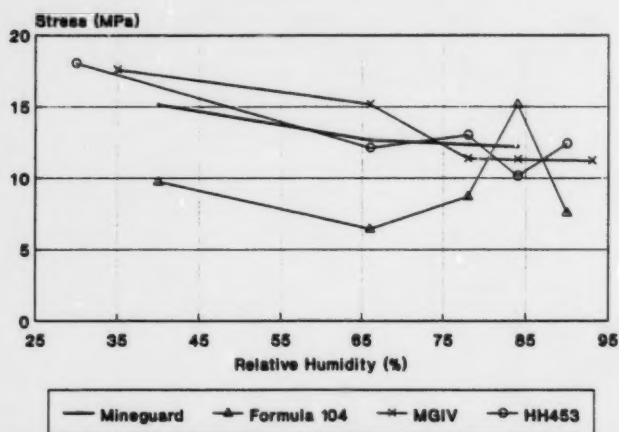


Figure 5a: Tensile Strength vs Humidity

a) adhesion strength of coated layers can be most significantly improved in high relative humidity conditions by using the Mineguard™ chemical variation HH453;

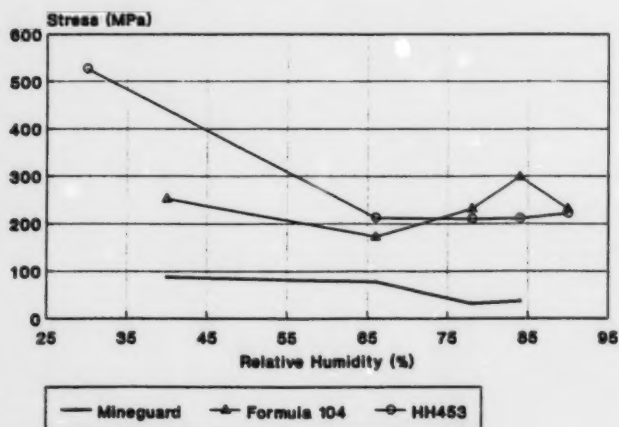


Figure 5b: Young's Modulus vs Humidity

b) Mineguard™ variation HH453 appears to exhibit improved material strength properties relative to all other variants, over the entire range of humidities tested;

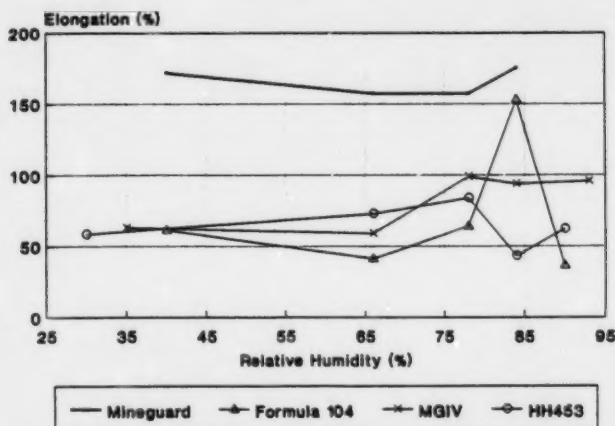


Figure 5c: % Elongation vs Humidity

c) all exhibited similar tensile strength behaviour to the original Mineguard™ formulation. Each was also demonstrated to exhibit significantly higher material stiffness (Young's Modulus), but decreased material elongation capability.

It would thus appear that the Mineguard™ HH453 variant offers significant resistance to the effects of high humidity and also provides consistently superior performance (compared to the original Mineguard™2 formulation) in all basic material property categories except elongation.

3.1.3 Review of Testing on Different Substrates:

An additional review was also undertaken to study the effect of spraying onto different rock (substrate) surfaces on development of adhesion strength

An additional series of perforated plate pull tests was initiated using the original Mineguard™2 formulation in which the polyurethane was sprayed onto natural rock slabs that were felt to be representative of a range of rock types. The typical rock materials used in these tests were sandstone, limestone and granite.

The results of these tests are summarized in Figure 6. It can

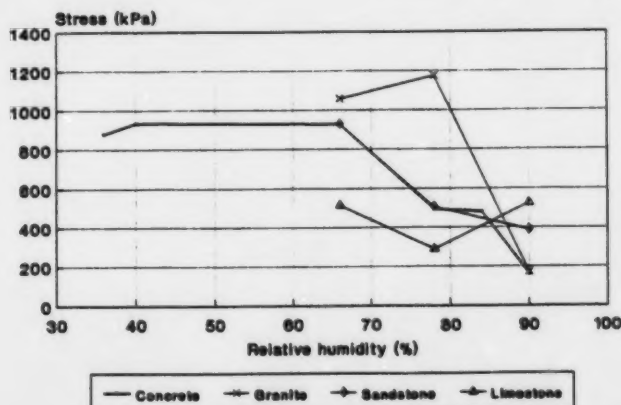


Figure 6: Adhesion Strength vs Humidity for Different Substrates

be seen from the plotted support resistance versus relative humidity results that Mineguard™2, when sprayed onto granite, exhibited the highest adhesion strength values; when sprayed onto limestone, exhibited the lowest results; and when sprayed onto sandstone, exhibited very similar results to those achieved using standard concrete slabs. The sandstone slabs appeared to have similar surface characteristics to those exhibited by the concrete slabs, while the limestone materials tended to have a waxy, clay-coated surface which would serve to reduce adhesion. The granite, alternately, was observed to exhibit rougher surface conditions and to have much greater surface strength than either of the other two types of rock.

These test results suggest that liner adhesion strength would likely vary underground according to encountered rock surface conditions, including strength and roughness. Test results obtained from solid pull plate tests have indicated that the strength of the lining itself was not significantly affected by variation in the type of substrate material.

3.2 Block Deformation Tests:

The second series of tests were implemented to provide answers for additional questions that developed during the underground test period. Primarily, it was thought necessary to determine, in a tangible way, the load resistance that could be generated by Mineguard™ linings in response to rockmass deformation and to determine the dominant mechanism by which such load could be generated.

The objective of block deformation tests was to approximate a specific situation that might occur underground in the excavation back within a blocky rockmass: that is, to duplicate a condition existing when a single "key type" block may be free to move relative to the blocks around it. In the worst case scenario this block would be free to slide under the influence of gravity along frictionless contacts.

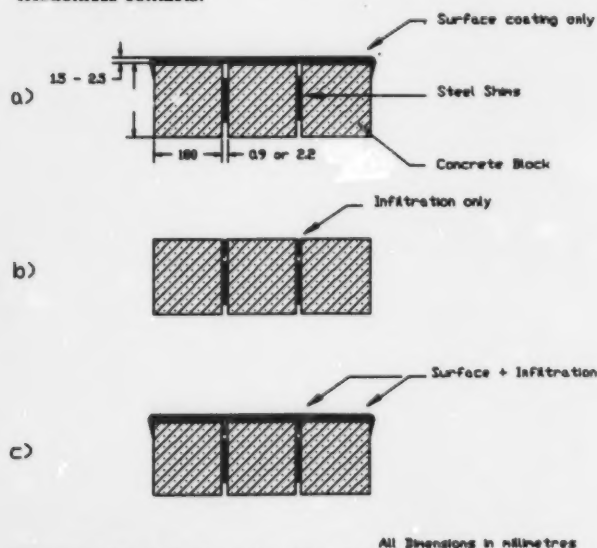


Figure 7: Block Deformation Test Configuration

Figure 7 shows a schematic of the basic experimental test set-up, which consisted of three 18 cm square concrete blocks which were held in place through their hollow centres by a tensioned bolt and nut arrangement. Of particular note is the inclusion of an evenly spaced gap between each adjacent block. This gap,

designed to be roughly analogous to a dilatant fracture, was created by inserting stacks of lightly oiled, flat steel shims between adjacent block surfaces. These shims not only provided an even gap space but also the capability to ensure that the middle block of the arrangement could move in a virtually frictionless manner past the two fixed outside blocks. Clearly, increasing or decreasing the number of shims in the stacks allowed the gap or "fracture" size to be changed.

The gaps were included between the blocks in an attempt to evaluate the relative contributions of fracture infiltration versus surface coating for provision of loose block restraint. Although the possibility of reinforcing the rockmass through fracture infiltration has been suggested in shotcrete applications^(6,7), its relative contribution has, to the authors' knowledge, never been quantified. This reinforcing effect could be even more significant in the case of polyurethane liner materials, due to the possibility of greater infiltration as a result of its lower application viscosity, combined with the contribution of surface layer stiffness, which is expected to be significantly less than that offered by shotcrete.

Three types of block deformation tests were conducted (depicted in Figures 7 a), b) and c)), and include:

- a) a surface only test, where only the surface of the concrete block assembly was coated with the Mineguard™ material (no fracture infiltration permitted)
- b) an infiltration only case, where only gap penetration was permitted (no surface coverage), and
- c) a combination of surface and infiltration cases.

A servocontrolled, electrohydraulic compression frame was used to load the centre block (after the bolt holding the blocks together was removed) and to measure both the deformation of the centre block relative to the fixed outside blocks and the resistance generated through the installed Mineguard™ layer (see Figure 8).

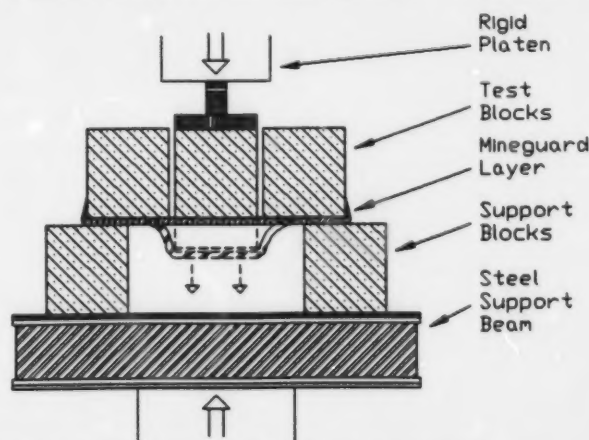


Figure 8: Block Deformation Testing Apparatus

Typical load-deformation test results are shown in Table 2 and illustrated in Figures 9 a) and b), representing graphs for block layouts having two different gap thicknesses (2.2 mm and 0.9 mm respectively).

The observations that can be made from these specific tests include:

Table 2: Typical Block Deformation Test Results

Test Type	Gap Thick. (mm)	Surface Coating Thickness (mm)	Fracture Penetration Depth (mm)	Peak Load (kN)	Peak Deformation (mm)	Total Deformation (mm)	Load per unit Length of Contact ³ (kN/m)
Surface	0.9	2.36	-----	1.73	6.50	44.5 ¹	4.55
	2.2	1.45	-----	1.10	6.50	39.0 ¹	2.89
Infiltration	0.9	-----	10.65	3.10	0.45	0.45 ²	8.16
	2.2	-----	32.30	13.0	2.00	2.00 ²	36.0
Surface + Infiltration	0.9	2.97	10.88	2.97 ^L 4.85 ^R	0.87 ^L 2.80 ^R	74.3	7.82 12.8
	2.2	N/A	35.30	16.0	2.20	11.5 ²	43.7

Notes:

L. - Left contact failure point, R. - Right contact failure point

1. - Final lining failure did not occur, 2. - Block failure resulted in twisting of apparatus, 3. - Based on 18 cm blocks

N/A - Not Available

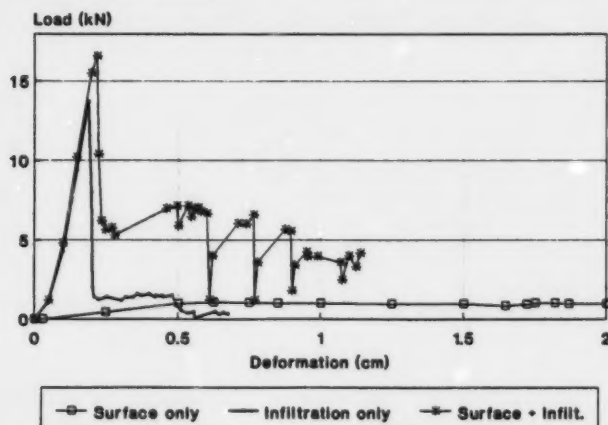


Figure 9a: Typical Block Deformation Test Results (2.2mm gap)

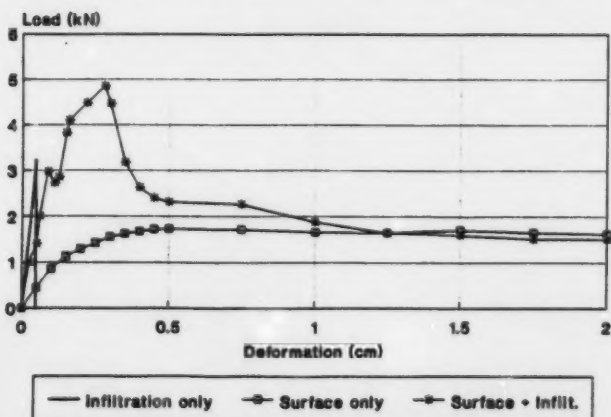


Figure 9b: Typical Block Deformation Test Results (0.9mm gap)

a) surface-only tests, in all cases, generated comparatively small load support resistance, while all tests that involved fracture

infiltration generated higher peak load resistance following relatively small deformations (generally less than 0.5 cm);

b) the initial stiffness response, for infiltration type tests, was similar regardless of gap thickness. However, the loading resistance generated improved with increased depth of penetration, corresponding to increased gap thickness;

c) the loading resistance generated in surface-only tests improved with increased surface coating thickness; and

d) for surface coated tests, layer coatings could sustain up to 7 cm of relative deformation between the blocks before final failure of the lining occurred.

In all the tests that involved fracture infiltration resistance, peak support resistance was limited by the strength of the concrete blocks, which broke directly adjacent to sites where the "fracture" was infiltrated. The breakage of the blocks terminated most tests immediately, since the blocks were no longer able to slide in a frictionless manner. The effect of friction can be seen in the surface plus infiltration test depicted in Figure 9a) in which, after concrete failure, peak support resistance fell to a residual load position which increased and decreased cyclically as the asperities along the broken face repeatedly sheared through. Worthy of particular note is the observation that, as the asperities sheared, the residual support condition was shown to develop in a manner identical to that observed to occur in the typical surface only test.

In one case, that being for the surface plus infiltration test depicted in Figure 9b), both infiltrated contacts were observed to fail in succession. Ultimately, after the second infiltration contact had failed, the block system exhibited characteristic load-deformation response typical of a surface-only test.

From these block test results, a first estimate can be made of the load support that might be generated by Mineguard™ linings in response to simple block deformation. In cases where at least some measure of fracture penetration could be assumed to occur, these resisting loads may be quite significant. It should be noted

that support contribution through fracture infiltration by polyurethane materials may become even more significant in terms of eventual support design, in that the majority of the sprayed material is used to coat over the rock surface while only a minority of the material would be used in penetrating the fractures.

4.0 CONCLUSIONS

In-situ installation tests have demonstrated that Mineguard™, in approximately 1.5 mm layers, can be rapidly and cost effectively applied. Average rates of application (including down time) were measured for Mineguard™ at an average of 1.5m²/min. at a cost approximating \$17.50/ m². Underground tests also indicated that Mineguard™, was not capable of withstanding a development blast without significant damage. However, these blast durability tests were conducted on Mineguard™ material exhibiting less than optimum physical strength characteristics, due likely to adverse effects created following exposure of Mineguard™ to elevated humidity levels.

Preliminary laboratory test results indicated that the deteriorating effects of high humidity may be mitigated by using a humidity resistant Mineguard™ formulation. From initial testing, at least one such Mineguard™ formulation has been shown to offer resistance to these effects.

The in-situ and laboratory tests conducted thus far suggests that thin Mineguard™ liners may offer competitive support capabilities compared to those offered by both shotcrete and bolt and screen support systems. This appears to be particularly valid when Mineguard™ is able to limit gravity - driven rock displacements by fracture infiltration.

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METHOD FOR COAL AGGLOMERATES STRENGTH DETERMINATION

K. Szymocha, W. Pawlak, J. Briker and L. Ignasiak
Alberta Research Council, Coal and Hydrocarbon Processing Department
P.O. Bag #1310, Devon, Alberta, Canada TOC 1E0

ABSTRACT

Currently, a variety of agglomeration techniques are used to improve the characteristics of the fine particles.

Depending on the method and conditions of the agglomeration process, the mechanical properties of agglomerates differ significantly. A good knowledge of these properties is important from the point of view of practical applications, such as agglomerates transportation and storage.

The existing ASTM test method D1937, which is designed for mass strength determination of carbon pellets, has been standardized since 1937. However, this method is characterized by low precision (repeatability $\pm 27\%$ and reproducibility $\pm 62\%$) and, as a result, is not very useful for studies on coal agglomeration or pelletization.

This paper describes the testing method used for determination of the mass strength of coal agglomerates and other pelletized materials. The results of these strength determination tests relate to the ability of the pellets to flow and to be stored in bulk handling systems.

The objective of this paper is to present the dynamic method and test procedure designed for mass strength determination.

INTRODUCTION

Ground materials exhibit characteristics which are very often undesirable. Such materials generate dust and are difficult to handle. Currently, a variety of agglomeration techniques are used to improve the characteristics of these fine particles. The mechanical properties of agglomerates differ significantly depending on the method and conditions of the agglomeration process. A good knowledge of these properties is important from the point of view of handling such as transportation and storage.

The strength of pellets can be determined by different methods. Depending on the test procedures, the strength is defined as:

- crushing strength
- tumbler strength
- mass compression strength

The test methods which were first developed for the determination of the physical properties gave only limited information of the handling behavior of the pellets.

Out of the existing test method, only a few tests are being considered as standards [1]. Constant efforts are being made to develop standard test methods and criteria for better characterization and description of the behavior of pelletized materials. In spite of the effort of ISO on an international level, a number of competing test methods are being employed in different countries [1].

The principles of the most popular methods are discussed below.

CRUSHING STRENGTH

By definition, pellet crushing strength is a force required to break an individual pellet. The crushing strength is measured by placing a pellet between two steel plates and applying pressure until the pellet is disintegrated. The major requirement of this standard test is that the increase in the load per unit time remains constant.

Usually ten or more pellets are crushed and the average force is reported. However, depending on the crushing strength and the accuracy required, it may sometimes be necessary to test as many as 100 individual pellets.

Benefits from using the crushing strength method are limited since the results cannot be directly related to mass strength. The crushing strength measurement is one of the oldest tests; however, proposals for standardization of this test procedure were made not long ago [2].

TUMBLER STRENGTH

The tumbler strength is another method for strength determination. The tumbler test measures the abrasion of the pellets in a rotating tumbler. A load of a representative sample of pellets having a size of 6.3 to 40 mm, is dried, weighed and then subjected to 200 revolutions at a drum speed of 25 ± 1 rpm.

After the tumbling is completed, the material is screened using 6.3 mm and 0.5 mm sieves. The plus 6.3 mm portion is designated as a tumbler index, while the minus 0.5 mm is designated as an abrasion index.

The method for determining the tumbler strength has been standardized since 1975 under the ISO Standard [3]. In the United States, the ASTM tumbler test (Designation E279-69) is used to measure the resistance of pellets to breakage upon abrasion and impact. Abrasion is measured by percentage of the -0.595 mm (-0.50 for ISO-3271 test method) fines produced during tumbling. Impact resistance is measured by the amount of plus 6.3 mm remaining after tumbling.

MASS STRENGTH

The existing ASTM test method D1937, which is designed for mass strength determination of carbon pellets, has been standardized since 1937. However, this method is characterized by low precision (repeatability +27% and reproducibility +62%).

Mass strength is measured by loading the pellets into a cylindrical column and applying plunger pressure until a bridge of material is formed. The air-driven apparatus is used for mass strength tests. The diameter of the stainless steel cylinder and plunger is 52 mm, while the length of the cylinder is 95 mm.

Mass strength is measured by the lowest number of kilograms required to produce a bridge of pressed carbon black in the cylinder. The formed bridge is inspected by opening a sliding door and observing the inside of the cylinder.

COMPACTION MECHANISMS

The generation of agglomerates or pellets can be carried out using different procedures. Most of the pelleted material is generated by the addition of specific binders. As a result, the products differ significantly with respect to physical properties (e.g. strength, plasticity, size, particle size distribution, shape). Depending on binder addition, agglomerates could be brittle or plastic [4], but have a similar compaction mechanisms. When spherical pellets are placed into a cylinder and force is applied (on the piston), a reduction in volume will occur as a result of compaction. Mechanisms of compaction of the pellets are shown in Figure 1. Three phases of the compaction process can be distinguished.

Phase I - At the beginning of the test, when the pressure is very low, dislocation of the pellets takes place. As a result, the agglomerates are transferred into void spaces. Rearrangement and more tight packing are observed (Figure 2). At this moment, the resistance of the agglomerate bed for contraction depends only on the friction coefficient between particles.

Phase II - An increasing pressure, when applied to brittle materials, results in pellet fractures and some limited rearrangement is seen. For plastic agglomerates, rather than fracturing, a plastic deformation with an increased area of inter-pellets contact is observed.

Phase III - When the majority of the agglomerates have been fractured, a further increase in pressure results in secondary crushing of the pellet fragments. For plastic agglomerates, with a further pressure growth, the resistance of the bed increases due to the few remaining voids. In Phase III, the broken or deformed agglomerates are not able to change their position. With a further increase in pressure, the piston achieves its final position and the apparent agglomerate bed density approaches the density of the agglomerates.

The contributions of the various mechanisms discussed depends on the physical properties of the pellets and on the speed of compaction. The major factors which may effect the test results are plasticity, particle size distribution, shape of pellets and others. The total displacement of the piston mainly depends on porosity of the bed which is a function of particle size distribution and shape of pellets.

According to Tanaka and Ouchiyaama [5], for monosized spherical pellets, the total theoretical bed contraction (piston displacement) should be in the order of 43%. In reality, due to the entrapment of gas or water in the bed when the voids become disconnected, and the difficulty to compact finely fragmented material as a result of higher strength of smaller fragments, the value of 43% is impossible to achieve.

COMPRESSIVE STRENGTH TEST METHOD DESCRIPTION

The proposed testing method is designed for the determination of the compressive mass strength of any pelletized material and, in particular, coal agglomerates. The method was developed as a result of Alberta Research Council laboratory testing and analysis of some compaction phenomena discussed by Pietsch [6]. A principal objective of the test is to determine the beginning of the distinct crushing process in the burden material. The proposed method utilizes dynamic compression of a sample of agglomerates. The test results are presented in the form of a compressive strength curve. The mass strength is defined on the basis of this curve.

The proposed apparatus is schematically shown in Figure 3. The apparatus consists of a mechanical or pneumatic load system, load indicator, displacement indicator and recorder for load and displacement registration. The load system is arranged in such a way that multiples of 1, 5 or 25 kg can be obtained. The pellets are placed in a cylinder which is 110 mm long and 52 mm in diameter.

In Figure 2, applied stress as a function of bed contraction is presented. The evaluation of the shape of the curve is a basis for qualitative analysis of the test. The mass strength is defined as the force (pressure exerted by piston) required to obtain a 10% contraction of the bed. It is assumed that at a bed contraction of 10% for brittle pellets, a crushing process is in progress and the weakest agglomerates have already been crushed. For a typical agglomerate sample, about 20% of the agglomerates can be crushed.

The 10% contraction value was established on the basis of the extensive laboratory testing performed with brittle and plastic pellets. The results of this mass strength determination test relate to the ability of the pellets to be stored or transported in bulk handling systems. The repeatability of the methods has been estimated to be +6% and reproducibility of 10% of test results.

RESULTS OF TESTING AND DISCUSSION

Mass strength determination experiments were carried out with the apparatus described above. The testing program was performed with coal agglomerates produced in a 250 kg/h continuous agglomeration unit. The agglomerates were characterized in terms of size, oil concentration and moisture content. In some cases, for completeness of evaluation, other factors were also considered. For example, the state of the agglomerates (drained or centrifuged) was responsible for the agglomerate quality.

In the first series of experiments, the repeatability has been evaluated. The size of the agglomerates used in these tests was in the range from 2.8 to 3.35 mm. Three different samples of agglomerates were investigated. The results of the experiments are shown in Figure 4. It has been found that repeatability of the tests was satisfactory.

The second series of experiments was carried out to analyze the impact of the agglomerates moisture content on the mass strength of the agglomerates. Results given in Figure 5 showed that as a result of reduction in moisture content a significant increase in the agglomerate strength was observed. A similar finding was observed and explained by Schubert [7].

The third series of tests was performed to study the influence of oxidation on agglomerate strength. In this test, agglomerates were exposed to air and samples were taken for mass strength determination. The results presented in Figure 6 show that oxidation improved the mass strength.

In the fourth series of experiments, the method was used for evaluation of agglomerate resistance during hydrotransportation. Pipelined agglomerates were taken from the pipeline at pre-determined distances and subjected to mass strength determination.

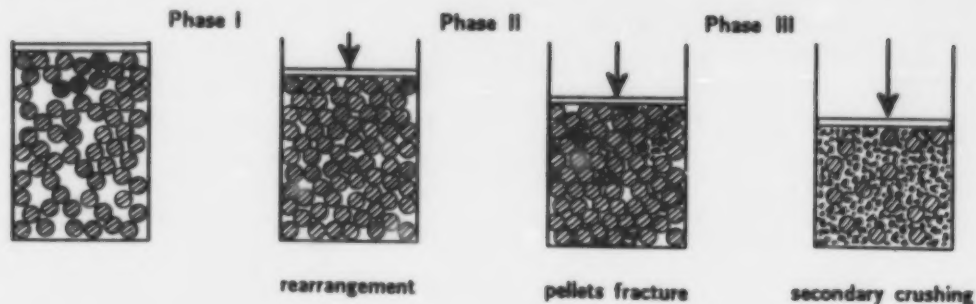
CONCLUSIONS

Results obtained showed that the proposed method for mass strength determination could be regarded as a very useful tool for strength measurements as shown by the tests performed under different conditions.

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A — brittle agglomerates



B — plastic agglomerates

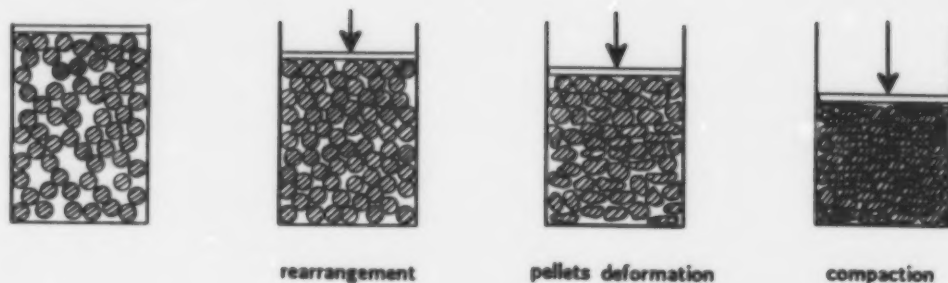


Figure 1. Mechanisms of compaction

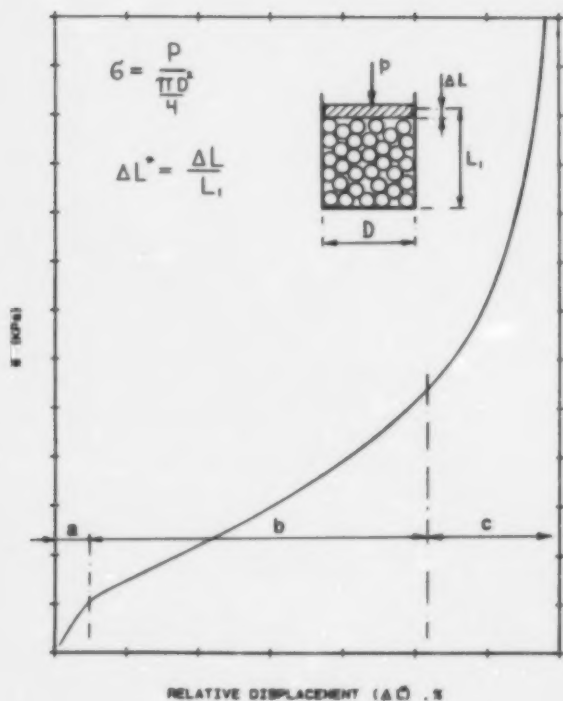


Figure 2. Strength of the bed of agglomerates as a function of contraction.

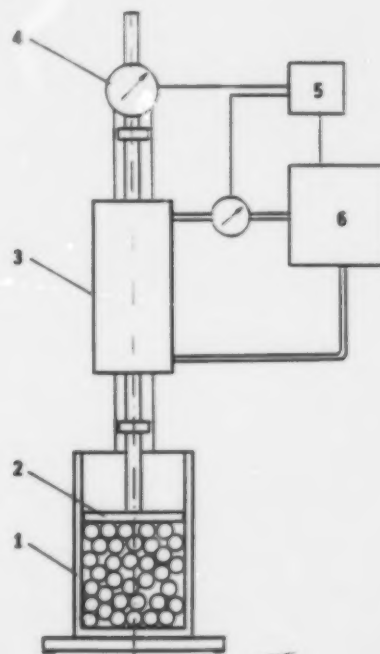


Figure 3. Apparatus for mass strength determination

1-cylinder, 2-piston, 3-pressure cylinder, 4-displacement indicator, 5-feed and control system, 6-recorder.

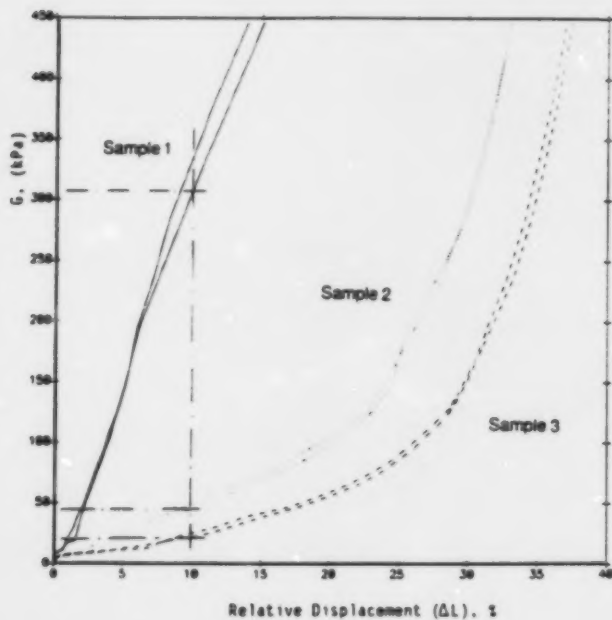


Figure 4. Repeatability of strength tests.

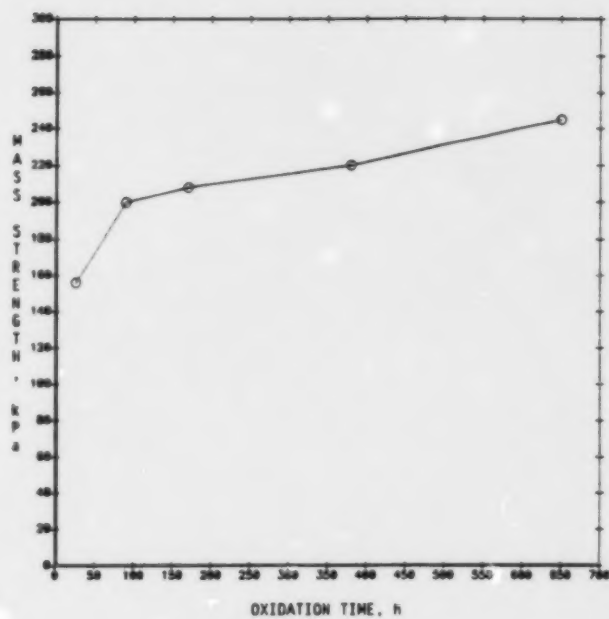


Figure 6. Agglomerate strength versus the oxidation time.

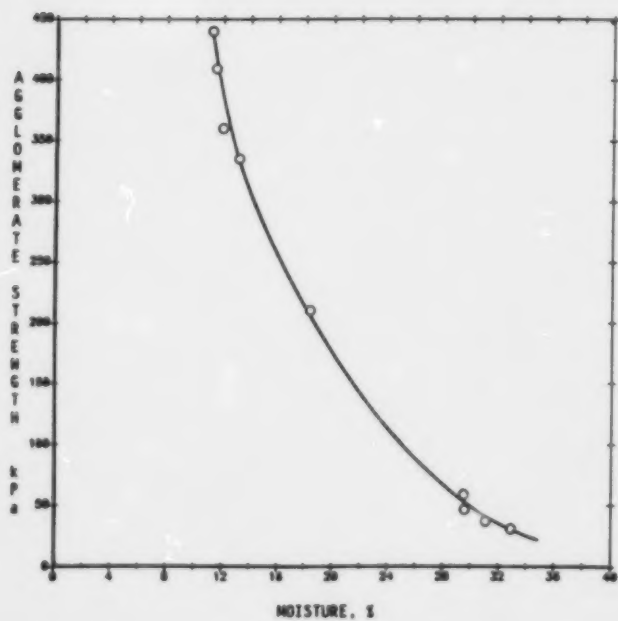


Figure 5. Impact of the moisture content on coal agglomerate strength.

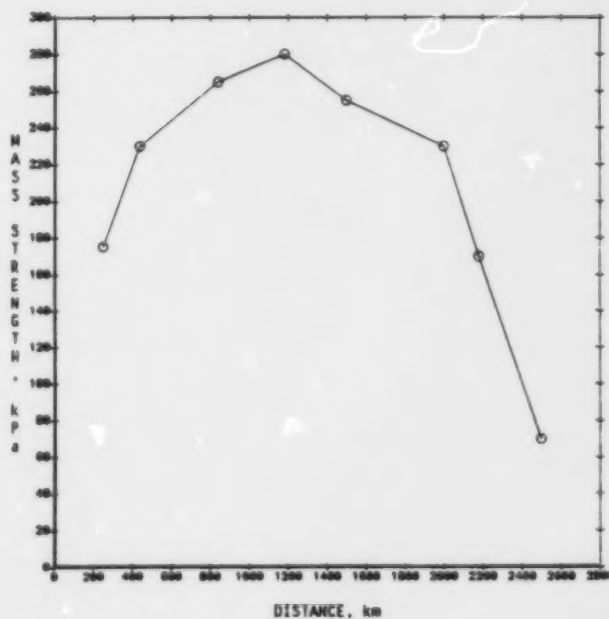


Figure 7. Agglomerate strength changes as a function of the pipelining distance.

SALT MECHANICS - SUPPORT AND SUBSIDENCE CONTROL IN SOFT ROCK MINES

E.M. De Souza & J.F. Archibald
Department of Mining Engineering
Queen's University

ABSTRACT

This paper describes the potential use of salt as backfill in underground soft rock mines, either to maintain local and regional ground stability during production or to limit creep deformation and control ground subsidence. The potential use of salt backfill as a feasible alternative to roof support is suggested and modified mine layouts which lead to improved extraction ratios and lower mining costs are evaluated.

1 INTRODUCTION

A large number of soft rock mines commonly use the room and pillar method of mining where relatively low extraction ratios are achieved. This restriction in the mining method and levels of extraction is due to the existing high stresses imposed by overburden, combined with the low strength characteristics of the rock strata and ore. Commonly associated adjacent water-bearing strata further requires that the stability of the pillars and rooms be controlled and maintained over the long term. The need for improved resource recovery, productivity and working conditions are some of the aspects of concern throughout the soft rock industry.

An alternative mining process, where increased extraction ratios are achieved, and structural failure and flooding are prevented by utilizing an artificial roof support system, would bring substantial benefits to the industry. This mining process would utilize waste salt as backfill material to support the roof while the ore is mined, thus ensuring the overall stability of the mine. The mining system incorporates a shortwall mining technique where the mined out area, behind the advancing face, is supported by stowing waste salt.

A detailed investigation of design properties of the backfill material is an essential part of the feasibility assessment of this mining concept. The suitability of waste salt for backfilling support in underground soft rock mines was determined by the investigation of its physico-mechanical behaviour, its strength, and its creep behaviour and failure properties. Laboratory tests were conducted to assess and predict the consolidation behaviour and strength development of the waste salt when used as a backfill, and to establish if it would be suitable to control underground deformations and surface subsidence.

Advanced computer simulation studies of innovative mining layouts have also been conducted to provide key answers with which the proposed methods and layouts could be safely and successfully introduced to the potash mining industry.

2 ENGINEERING PROPERTIES OF WASTE SALT BACKFILL

An extensive laboratory testing program was undertaken to evaluate the engineering properties of waste salt for backfilling underground potash mines in high extraction mining methods and layouts.

The program was designed to study the relationships between granulometry, moisture content, density and consolidation pressure on the creep and consolidation behaviour of waste salt when tested under static and quasi-static conditions, and to study the strength characteristics of the backfill under conditions as anticipated

underground. The study objectives were to assess and predict the time dependent creep-consolidation behaviour and strength development of the waste salt when used as a backfill, and to establish if it would be suitable to control underground deformations and surface subsidence. This study would allow us to introduce new concepts of potash mining where high extraction ratios are involved.

The laboratory tests performed for the evaluation of deformation and strength parameters of the backfill included static and quasi-static consolidation tests, standard and step-loading creep tests, uniaxial and triaxial compression tests, and shear tests. Modelling and parameter investigation of the backfill material nonlinear behaviour was performed using elastoplastic and viscoplastic rheological models. A series of characterization tests were also performed to provide an indication of the physico-mineralogical characteristics of the backfill material, including mineralogy, specific gravity, moisture content, granulometry, standard dynamic compaction tests and percolation tests.

2.1 Backfill Characterisation

The material constituting the backfill consisted of 85 to 88 percent Halite (NaCl) and Sylvite (KCl), at a 2:1 ratio, and 8 to 10 percent clay minerals. The specific gravity of the waste salt was measured as 2.175 g/cm³; the bulk density, as received, as 1.52 g/cm³; and the porosity calculated as 30%. The moisture content at which this material would reach full saturation was determined as 19.7%. It was noted however that because of the granular nature and permeability characteristics of this material, a stowed, coarse backfill would never maintain a saturated state; the brine present in the material drains freely to very low values of final moisture content, and backfill liquefaction should not occur.

Grain size distributions indicated that the received material had an effective grain size of 0.64 mm and a Coefficient of Uniformity of 7.8, indicating a graded material. Three new materials were constructed; the corresponding Coefficients of Uniformity (d_{60}/d_{10}) for each were 10.83, 7.67 and 3.57, the effective grain sizes (d_{10}) were 0.13 mm, 0.086 mm and 0.060 mm, and the 50% sizes passing (d_{50}) were 1.02 mm, 0.50 mm and 0.175 mm, respectively. The first material indicated better gradation than the as-received material, and the third showed a marked increase in uniformity.

A relationship between the maximum dry density of compacted waste salt and moisture content was established for the three materials to determine the stowing characteristics of the backfill that should be used to minimize the amount of roof deformation and subsidence. It was found that higher maximum dry densities are

achieved for coarser, graded materials, and that backfill materials, constructed of waste salt, should be stowed during mining at ideal moisture contents between 8 and 10%. Proctor tests indicated a maximum dry density of 1.80 g/cm³ when using a material at a 50% size of 1.02 mm and with an initial moisture content of 8.5%.

Because of concern about the amount of excess brine that will have to be coped with during underground operations, the capability of the material to retain brine was also evaluated. It was noted that brine starts flowing from the material at moisture contents greater than 11% for the coarser material; the finer, more uniform materials could retain brine to 13 and 15% moisture. The brine, however, would drain to much lower moisture contents after standing over a period of time. The results indicated that saturation of the coarse product and failure due to the presence of large amounts of brine in the early stages of material consolidation would mostly likely not occur. The results also confirmed that a moisture content between 8 and 10%, with these materials, would provide ideal conditions for stowing the waste salt; mixing brine in excess of this value of moisture content would simply drain through the backfill, probably causing operational problems rather than giving any benefits. This indicates that a drainage system will have to be designed to remove any brine draining from the backfill; with drainage volumes reaching 135 litres per cubic meter of backfill.

Percolation tests showed that an increase in backfill fines drastically reduces the percolation characteristics of the product. Because the material would be stowed at low moisture contents, as suggested, there would be no immediate concern about brine saturation. However, in areas of probable brine breakthroughs, the placement of coarser products would be preferable.

2.2 Backfill Strength

Strength testing indicated that waste salt can form a strong backfill following 27% creep-consolidation. Considering that stowing practices cannot place the backfill tight against the roof of panels, backfill would be capable of reducing potential subsidence due to full extraction mining by approximately 64%. Ultimate deformation is dependent upon granulometry, initial moisture content, stowing density, and time between mining and backfilling. The backfill mechanical behaviour and strength development has been characterized during the different phases of mining:

Backfill Behaviour and Strength at Placement - During backfill placement the presence of brine has a marked effect on the consolidation behaviour of the backfill. It is apparent that brine has a lubricating effect at low stress, facilitating backfill consolidation and closer packing of the salt grains. A maximum consolidation of 35% was achieved for a stress of 1.43 MPa. Maximum compaction, maximum density, and minimum porosity are achieved at placement for coarser, graded materials, and at ideal initial moisture contents between 8% and 10%; allowing the backfill to develop in situ equivalent loads under minimum consolidation. Mixing brine in excess of this range of moisture content would drain through the backfill, probably causing operational problems rather than giving any benefits. This initial moisture also provides a more consistent material which could be stowed tighter against the panel roof.

Brine drainage analysis indicated that facilities would have to be designed to handle total brine volumes of 135 litres per cubic meter of backfill, more than 50% of which would be drained and needed to be handled within the first few hours of placement, and up to 80% when initial load from room closure is transferred to the backfill.

Following placement, and during the initial stages of load uptake due to room closure, only a weak, friable backfill is formed, exhibiting minimum unconfined strength. Only minimum grain interlock and bond strength is created at this stage. It is sufficient to aggregate the material but still not adequate to allow for the backfill to carry extended load. A maximum uniaxial compressive strength of 0.71 MPa was obtained for a material consolidated by a maximum stress of 1.41 MPa and with an initial moisture content of 5%. A maximum triaxial compression strength of 1.74 MPa was obtained using a confining stress of 0.34 MPa. Failure of this material under unconfined conditions is characterized by the development of a shear surface, although large lateral deformations and bulging occur prior to backfill

destruction. Also, at this stage, backfill confined failure occurs preferentially through a mechanism of buckling and collapse, throughout the entire material, rather than by localized shear. This failure mechanism indicates the importance of backfill confinement during the stowing stage, otherwise it would simply be squeezed out of the panels rather than achieve solid consolidation, and its use for long term ground support and subsidence control would be considerably reduced.

Backfill Behaviour and Strength at Stages of Load Development and Consolidation - Consolidation tests showed that coarser, well-graded, crushed salt materials, rather than finer, more uniform materials, provide superior backfill, as they characteristically achieve a higher density at placement, develop in situ equivalent loads under the smallest amount of consolidation and develop higher strength. Consolidations of 24%, 26.2% and 32.96%, for a stress of 20.68 MPa, were achieved for the coarse, medium and fine materials, respectively. Also, because coarser, well graded materials can attain maximum placement density at smaller initial moisture contents, the amount of excess brine to be handled during consolidation will be much lower than for finer, more uniform materials, thus reducing the drainage facility requirements.

Strength test results on specimens prepared under high stress quasi-static consolidation, which represents the advanced stages of high load transfer from room closure, indicated that failure occurs through shearing mechanisms. A maximum uniaxial compressive strength of 4.5 MPa was obtained for a material creep-consolidated by a maximum stress of 20.68 MPa and a maximum triaxial compression strength of 7.79 MPa was obtained using a confining stress of 1.38 MPa. Comparison with results from low stress consolidated backfills indicates that there is a change in backfill strength behaviour as it is increasingly consolidated. At placement and during the first stages of consolidation under low stress, there is no substantial gain in strength, and the backfill failure mechanism is preferentially one of bulging and collapse. With further consolidation, under stresses equivalent to in situ values, the backfill failure mechanism becomes one of shear failure, characterized by its developed cohesion (45 KPa) and angle of internal friction (33°). The increase in strength with consolidation is explained by mechanisms of grain bonding which occur during consolidation.

Direct shear tests also confirmed that a build up in backfill strength occurs through the development of friction and cohesion between its particles as a function of applied stress. The tests indicated that the angle of internal friction is not dependent upon the initial moisture content, but that the apparent cohesion, resulting from surface tension in the brine held at grain contacts within the material, is maximum at an initial moisture content of 8% and zero for dry conditions and moisture above 10%. Those results confirmed that the backfill should be placed in situ at an initial moisture content of approximately 8% in order to provide an improved backfill in terms of strength during load development. At moisture contents above 10%, under undrained conditions, an inadequate, cohesionless material would be produced.

A nonlinear consolidation-stress relationship was found to exist for the salt material, which allows for the prediction of backfill consolidation at any given in situ stress condition: $\epsilon_c = \sigma / (E_c + E_0 \sigma)$, where ϵ_c is the consolidation strain (cm/cm), σ is the consolidation stress (MPa), and E_c (MPa) and E_0 (dimensionless) are parameters defining the stress dependent behaviour of the modulus of deformation. Using estimated values of E_c and E_0 , the tangent modulus of deformation for the material is defined by: $E_t = E_c + E_0 \sigma = 19.638 + 3.219\sigma$. This equation can be used to predict backfill consolidation at any given in situ stress condition.

An expression was also developed to estimate the ultimate room closure due to consolidation of the backfill: $\epsilon_r = \epsilon_c + \epsilon_v$, where ϵ_r is the total room strain for the backfill to achieve full consolidation, ϵ_c is the volumetric room strain related to incompleteness of backfilling and ϵ_v is the total volumetric backfill strain required for full consolidation. A backfill testing program was established for an underground room and pillar panel and the following predictions were derived: Each room, before backfilling, was 3.35 m high and 4.0 m wide, and the gap between the panel roof and the backfill was 0.305 m. ϵ_c is thus estimated as 0.091. The backfill was placed dry,

at an estimated maximum density of 1.5 g/cm³ and initial porosity of 31%. At full in situ load transfer due to room closure, consolidation was predicted to reach 26%. For maximum consolidation, ϵ_c is thus estimated as 0.26 and ϵ_s as 0.351. This indicates that full consolidation is achieved when the room volume is reduced by approximately 35%, or 4.69 cubic metres per unit length of backfilled room (m³/m). The time required for full backfill consolidation can also be estimated. Considering measured vertical and horizontal room closure rates of 1.51 cm/100 days and 1.24 cm/100 days, respectively, the volume reduction per unit length of backfilled room, V , (cm³/cm), can be written as a function of time, t , in days, as: $V = t(10.194 - 0.000187t)$. The backfill will be fully consolidated when the room volume per unit length is reduced by approximately 46,900 cm³/cm. Solving for the equation above, it will take approximately 14 years for the backfill to be fully consolidated and to provide full panel support. The total room vertical closure which will occur until full backfill consolidation results was estimated as 0.77 m, which indicates that potential subsidence caused by mining can be reduced by 77% using dry waste salt backfill. If the backfill had been placed at an initial moisture content of 8%, full consolidation occurs after 13 years, a reduction of approximately 1 year. If the backfill is stowed tight against the panel roof, the time would be further reduced to approximately 9.2 years, which illustrates the importance of operation control during placement.

For high extraction mine methods, where the backfill is placed in large block spans, only the vertical closure plays an effective role in the consolidation of the backfill. For total subsidence prediction, an additional component of strain, ϵ_v , which considers vertical strain occurring immediately after mining and prior to backfilling, has to be included in the equation, giving: $\epsilon_t = \epsilon_i + \epsilon_v + \epsilon_c$, where parameters are now defined as: ϵ_i is the total vertical room strain for the backfill to achieve full consolidation, ϵ_v is the vertical room strain related to delay in backfilling, ϵ_c is the vertical room strain related to incompleteness of backfilling and ϵ_s is the total vertical backfill strain required for full consolidation. For high extraction mine methods, where much higher spans are involved and high closure rates occur, backfilling will proceed immediately after mining, and the consolidation times will be drastically reduced, to months.

Backfill Behaviour and Strength at Stages of Full Load and Creep - Under creep conditions and for applied stresses equivalent to in situ values, up to 99% reduction in initial moisture content may occur, producing final moisture contents between 0.2% to 1%. High increases in density are also predicted, to give relatively high final densities, corresponding to 96 to 98% the material true solids density. Although the initial density at placement is a function of the initial moisture content, final densities seem to be more dependent on the time the backfill is subjected to load and on the magnitude of the applied stress. Decrease in porosity from 83% to 90% of the initial value will produce final values in the range of 1.5% to 4.0%, indicating the potential use of backfill for bulkheads to isolate areas of brine inflow. Convergence may reach 27% to 40% the original mine opening height in a full extraction mining method using backfill. This indicates a potential reduction in surface subsidence due to mining of up to 73% for ideal backfill placement.

After creep-consolidation, the backfill strength response under confined and unconfined conditions is one of controlled failure through shearing. When subjected to static loads over long periods of time, the backfill demonstrated large increases in strength, suggesting that time under load has a marked effect on the development of backfill strength. Test results indicated that salt backfill materials, after long term creep under high static stresses, develop relatively high strengths and behave in a similar fashion to the original salt rock formation. This strength-time response mechanism has several practical implications, as it allows for effective planning of mining strategy, sequencing and most importantly, rate of mining. In order to achieve high extraction ratios it also permits accurate prediction of long term strata behaviour and failure as the behaviour of the backfilled panel would be equivalent to that of the salt formations under the same stress field.

The mechanisms involved in high stress creep-consolidation and backfill strength development are independent of the initial moisture content, as brine drains to very low moisture contents at the very

early stages of consolidation. They are more a function of rate and magnitude of load, and time under sustained load. The period of time during which the backfill is subject to load appears to have a greater effect on strength development than the magnitude of load itself. At full load the backfill will attain low moisture, high density, low porosity and high modulus characteristics. At this stage, the backfill materials behave in similar fashion to the rock from which they were originated, thus indicating their value as a mechanism of ground control.

Backfill thin slice analysis indicated that a mechanism of grain growth occurs during creep. This grain growth mechanism, which occurs due to grain bonding and matrix cementation, generates very intimate grain contacts, and is caused by recrystallization of concentrated brine when subjected to high stress. Strength is thus not only formed by mechanical interlock of grains but is also developed by recrystallization of concentrated brine and by intergranular bond growth. For cases where the backfill is subjected to high static stresses over long periods of time, a solid matrix of salt is formed, to exhibit strengths similar to that of salt rock. Under this condition, most of the brine is expelled and reprecipitation of salt from the brine remaining within the pore spaces follows as a function of stress and time. This causes a condensed agglomeration of the grains, producing a matrix of solid salt with similar characteristics to natural rock salt. The change in intergranular characteristics explains the time dependent strength growth mechanism, and the change in backfill failure behaviour, from buckling to shear, as the consolidation stress and time under load are increased.

A practical constitutive relation was developed to describe the creep behaviour of a backfill acted on by mine stresses and over long term conditions as encountered underground. The creep law is a semi-empirical exponential-time law; it is based on the principles governed by first order kinetics, and has the following form:

$$\epsilon = \epsilon_i + \epsilon_v + \frac{\epsilon_c}{C} [1 - \exp(-B C t)] \quad (1)$$

where ϵ is the total strain, ϵ_i is the initial strain related to the elastic response due to loading, ϵ_v is the steady-state creep rate, ϵ_c is the asymptotic transient strain, t is the time, and B and C are fitting constants. The instantaneous strain, ϵ_t , is non-linearly dependent on the applied stress, and their relationship has been developed before. The steady-state creep rate, ϵ_v , is given as $\epsilon_v = A\sigma^n$, where σ is the applied stress, A a fitting parameter, and n a fitting exponent. Parametric analysis using experimental creep data on backfill, resulted in the following parameter values: $E_s = 19.638$ MPa, $E_c = 3.219$, $A = 0.695 \times 10^{-7}$ MPa⁻ⁿ sec⁻¹, $n = 0.12$, $E_v = 1.06 \times 10^{-3}$, $B = 159$ and $C = 5.1 \times 10^{-4}$ sec⁻¹. This relationship can be used to describe backfill creep over the range of stresses expected in a full extraction mining situation, and to predict long term performance of salt backfill.

Using expected mine closure rates during high extraction mining, and using backfill strains estimated by the creep relationships above, total room closures due to consolidation and creep of the backfill in total extraction mining strategies can be estimated for different periods of time. The following relationship is used to estimate the total strain necessary for the backfill to achieve full strength: $\epsilon_t = \epsilon_i + \epsilon_v + \epsilon_c + \epsilon_s$. The parameters are now defined as: ϵ_i is the total vertical room strain at which backfill achieves full strength, ϵ_v is the vertical room strain related to delay in backfilling, ϵ_c is the vertical room strain related to incompleteness of backfilling, ϵ_s is the total vertical backfill strain required for full consolidation, and ϵ_t is the time dependent backfill creep which is the mechanism responsible for backfill strength development.

The strain due to consolidation, and the strain due to the gap left between the roof and the top of the backfill, occur during a time which is more dependent on the mine closure rate. Once those strains occur, the backfill will be fully consolidated and further strain is governed by creep behaviour of the material as described by the second and third terms of the creep equation. In general, mining should be immediately followed by backfilling. However, considering that a one to two day delay in backfilling would normally occur after extraction, an average value of 0.004 is given to ϵ_v . A value of 0.09 is given to ϵ_i to represent the free deformation occurring due to estimated gap of 0.3 m between the roof and the top of the backfill.

The strain due to full consolidation is given by the first term of the creep equation as 0.24. As determined by the series of laboratory tests, after approximately 27% creep-consolidation has occurred, the backfill attains approximately 98% of the density of solid salt rock and very low porosity. It also achieves high strength and full room support. This gives a value of 0.03 for ϵ_c . Thus, the total vertical room strain, ϵ_v , for the backfill to achieve full strength is estimated as 0.36.

Based on measured and estimated mine closure rates, and considering conditions of full room extraction, it would take approximately 1 month for the roof to touch the top of the backfill and approximately 5 months for the backfill to be consolidated. The additional time required for strength development due to creep strain, according to the creep equation, is approximately 1 month. For those practical applications, only the secondary creep was used in calculating the time required for backfill strength development, because of the comparatively slower loading of the backfill due to room closure. Thus, after approximately 7 months, 36% creep-consolidation is predicted, which corresponds to 27% backfill strain, plus 9% of free deformation due to an estimated gap of 0.3 m between the roof and the top of the backfill. This is the approximate time necessary for the backfill to achieve full strength. This simplified sequence of calculations illustrates how time periods for backfill full load support can be estimated; those time spans are essential for designing panel sizes, mining rates and mining sequence.

3 ASSESSMENT OF A HIGH EXTRACTION MINING METHOD

A study to provide realistic and practical engineering solutions for the successful introduction of alternate layouts to efficiently and safely mine soft rock underground has been developed. The most important aspect of the study was careful consideration with respect to mine integrity and safety. From a review of mining methods used in soft rock producing countries around the world, shortwall mining was found to satisfy the engineering requirements for the successful application of a high extraction method.

A computer code capable of realistically simulating the designed layouts and strategies was calibrated using field and laboratory investigations and subsequent mine simulation was performed. Several possible scenarios of mine design innovation were analyzed and proposed.

3.1 Shortwall Mining

The most favourable method for mining flat lying and continuous soft rock deposits, which would ensure the integrity of a mine and eliminate the risks of brine inflow, is the shortwall retreat mining method incorporating backfill and yielding pillars. The use of backfill, associated with the shortwall method, is vital to safely introduce this alternative layout underground. The application of yielding pillars is recommended to ensure stable roof conditions at the working face. Beneficial gains over current methods in terms of improved safety, productivity and ore recovery were found possible when using high extraction mining systems.

In the design of a shortwall mining system, it is believed that all the concepts that have been proven effective in conventional soft rock room and pillar mining should be employed or adapted underground. A suggested panel layout for initial field trials is suggested, as illustrated in Figure 1. The system incorporates backfill and thin yielding pillars for support. Using this arrangement, the yielding pillars would serve to provide support close to the mining face and the backfill would provide the required regional support. Using such layouts, improved panel recoveries would be achievable.

3.2 Computer Modelling of Shortwall Mining

A two-dimensional and axisymmetric, non linear, transient, thermo-stress finite element code, named VISCOT¹, designed to determine the viscoelastic, viscoplastic or elastoplastic deformation of a rock mass due to mechanical and thermal loading, was used in the modelling studies. The code can perform plane stress, plane strain and axisymmetric analysis. Isoparametric quadrilateral elements are employed. The elements used include four-noded linear quadrilateral elements and eight-noded quadratic serendipity elements. Isothermal,

steady state thermal, or transient thermal stress analysis can be performed using the code. The following constitutive models are available within the code: linearly elastic; non linear (stress dependent) elastic; viscoelastic (transient or steady state, time dependent and strain hardening); and viscoplastic (Tresca, Von Mises, Drucker-Prager, and Mohr-Coulomb criteria, strain hardening, associated plastic flow).

Typical applications of the code include model studies during the excavation, operation, and long-term post-closure stages of excavations in a rock mass such as salt, in which significant time-dependent non linear deformations are expected to occur. The code can be used to simulate the mechanical behaviour, the induced displacements, and the state of stress of a rock mass or underground structures during the excavation process. Evaluation of room closure rates, stability, floor heave, overall room integrity, the state of stress and long term deformation behaviour of salt backfill materials placed underground may be made using this code.

3.2.1 Code Calibration

In order to determine the technical feasibility of the high extraction mining concept it is first necessary to model currently used and well established mining methods to permit calibration of the computer code. Successful calibration then permits evaluation of unknown mine layouts and mining strategies.

Calibration of the finite element code was based on modelling the layout established in a mining panel with respect to known conditions underground as established by a detailed field instrumentation program. An iterative procedure was employed for calibration of the code. The method of calibration included performing a number of computer runs to first examine the sensitivity of the analysis to each parameter describing a particular material, to examine the sensitivity of the model to assumptions made to mathematically describe material behaviour, and to determine which parameters should be more carefully evaluated during the calibration of the code according to known field measurements. This would not only assure that a direct correspondence between laboratory determinations and field observations is achieved, but establishes if the assumptions made to derive the analytical models and constitutive relations and if the concepts established from laboratory experiments are accurate and valid to describe the rock masses. Based on this calibration procedure, preliminary field predictions were made and the inconsistencies existing between field measurements and model predictions were improved at each run until favourable agreement and sufficient accuracy was achieved.

The correlation procedure included comparison of model results and measured stresses and deformations in the mine panel. Consistent and accurate correlation between the predicted material behaviour using the theoretical models used in the computer code, the material parameters determined in laboratory tests, and the direct field measurements was achieved. Also the predicted strata behaviour and failure conditions were in close correspondence with those observed underground. Calibration results indicated that the calibrated code could be confidently used for evaluation of a full-scale high extraction underground operation.

3.2.2 Simulation of Alternative Mine Layouts

The success of calibration lead to confident use of the code for extrapolation from the model to untested layouts and mining strategies. A number of models were designed and tested to evaluate whether such layouts could be practically and safely introduced underground, and to study the applicability of several possible scenarios of mine design innovation. The development of these models and the results achieved are discussed.

Simulation of a Single Panel Mined Using the Shortwall Method - Modelling of the shortwall mining method with backfill was performed in order to evaluate the general ground conditions developed during mining. Modelling performed was of a cross-section through an active face with a backfilled gob in a potash mine. The finite element grid used in the analysis is illustrated in Figure 2. Eight-noded quadratic serendipity elements were used to accurately

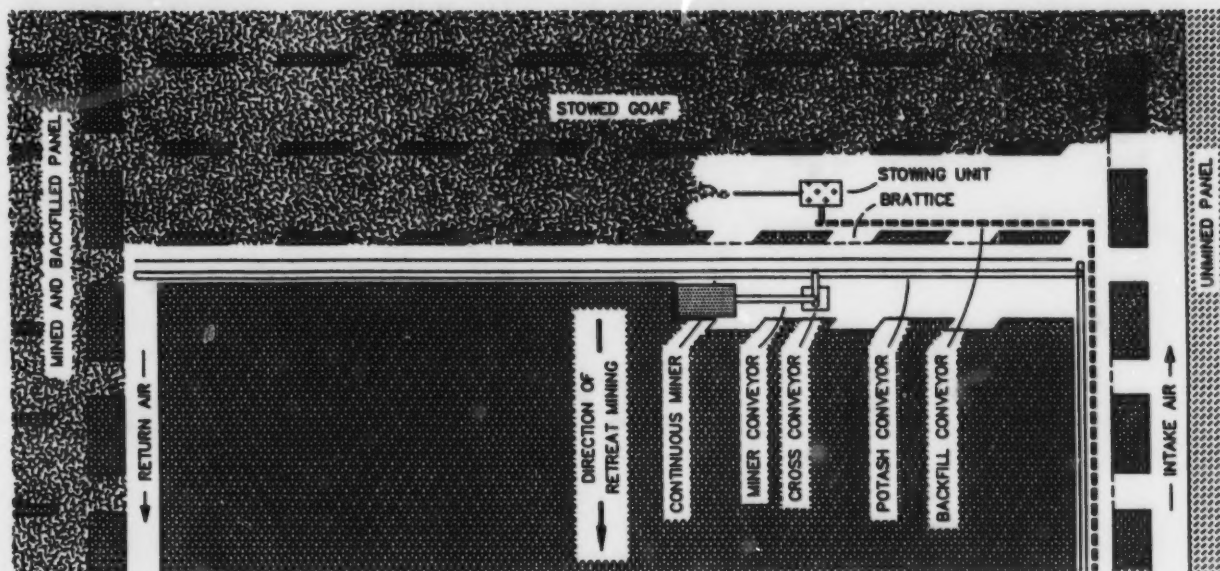


Figure 1 - Shortwall Mining

describe the non linear behaviour of the rock masses. A total of 168 elements, defined by a total of 563 nodes, were used in the discretization procedure, which was designed to model the actual geological setting existing underground. The lateral boundaries were located 180 metres in the potash seam in advance of the last simulated face in the model and 180 metres back from a 169 metre backfilled gob section, established at the end of the simulation runs. The working face was 11 metres wide.

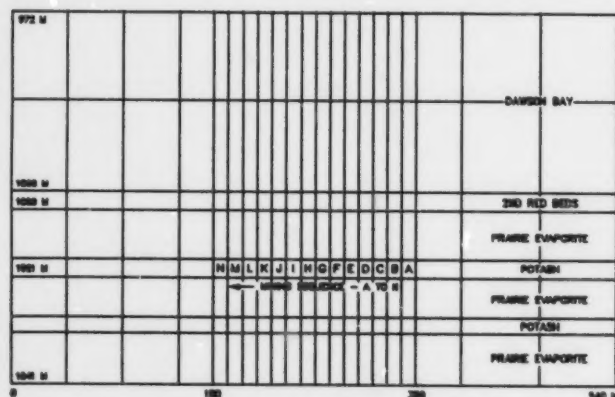


Figure 2 - Finite Element Mesh

The analysis was performed to evaluate stress and strain conditions developed along a shortwall panel longitudinal section during mining of the potash seam. The simulation analysis included determination of the node loads equivalent to the stress condition existing prior to retreat to a new working face, and then relaxation of these loads in a prescribed time interval to simulate retreat of the face by two miner passes. The non linear stress-strain calculations were performed in real time, taking into account existing field conditions due to room excavation and due to backfilling. Fourteen runs were performed to simulate retreat of the face from the beginning of the panel, 180 metres into the potash seam. Each run simulated two passes of the miner, each pass being 5.5 metres wide (miner width). After each two pass extraction, a yielding pillar 2 metres wide is left, and the previous excavation is backfilled. A slightly conservative approach was adopted in not including these

yielding pillars, used to provide local roof support close to the face, in the model. Each run simulated a 51.2 hour cycle. This time of retreating 11 metres of face was estimated based on a face 384 metres long and on a miner advance rate of 15 metres/hour. A mining simulation schedule totalling 716.8 hours was thus established to evaluate ground reactions to active mining conditions in which the face would have retreated from the beginning of the panel a distance of 180 metres into the potash seam.

Figure 3 shows the stress conditions along the mine level, in the salt back, and in the limestone formation, at the time when the last face in the model is developed (element N in Figure 2), and at different times of previous face retreat and backfilling. The conditions encountered at the seam level are detailed. From a zero stress condition at the face, the stress increased to a maximum value in the abutment zone beyond which it decreased again and ultimately attained the pre-mining stress level. The peak stress reached magnitudes 1.36 times the original virgin stress level. The deflected position of the peak stress from the face into the unmined seam was due to yield and fracture of the immediate face. The depth from the face at which undisturbed stress conditions were attained was estimated at about 130 metres.

The immediate backfilled gob did not carry any superincumbent load at placement time. Alternatively, stress conditions had been developed in the mid section of the gob area, which had been mined and backfilled 358.4 hours prior to the existing face. These conditions indicated that the backfill already carried part of the superincumbent load and provided partial support. Further back into the gob area, the backfill, at 716.8 hours after placement, had developed higher support capability. At this time, however, full load development had not yet been achieved. As will be shown later, it takes approximately 7920 hours for the backfill to provide high support to the back for a panel 384 metres wide.

The induced stress conditions existing in the overlying salt and limestone formations are also shown in Figure 3. The abutment stresses affect both formations to a small extent. Also, an arching effect, resulting from a zone of stress relief, was noted to develop. This zone of decreasing stress concentration extended from the abutment area onto the gob section, to a depth where the back was still unsupported by the backfill. The influence of such 'disturbed zones' was noted to decrease as the backfill developed load.

Figure 4 shows the strain profile occurring across the backfilled room longitudinal section. Approximately 10% maximum strain occurred in the mid panel section before the backfill started to develop load. The strain rate decreased with time as the backfill became stronger. At a time 358 hours after mining, the mid section

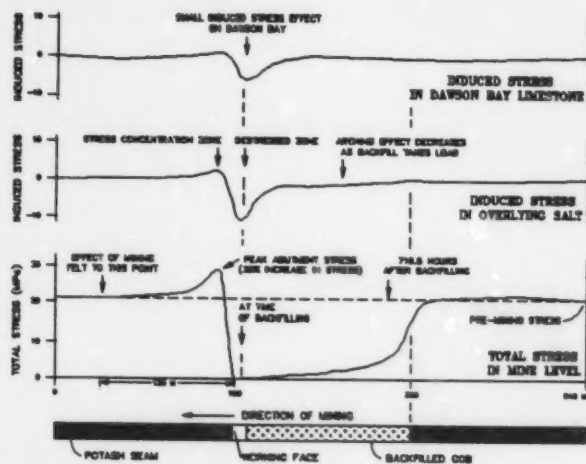


Figure 3 - Stress Profile Around a Shortwall Face

of the panel had reached 12% strain.

These results indicate that the stress and strain conditions developing from shortwall mining do not create extreme conditions which would pose a risk to overall mine integrity. Backfill proved to be effective in controlling ground closure in a relatively short period of time after mining. These results must, however, be verified through field trials.

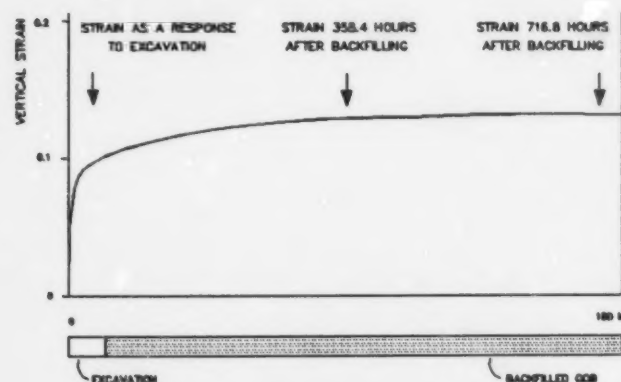


Figure 4 - Strain Profile Around a Shortwall Face

Simulation of a Sequential Retreat Strategy to Mine a Block of Shortwall Panels - Modelling performed was of a cross-section through two adjacent panels in order to evaluate the general interaction between the backfill and surrounding strata, and to estimate the time required for full backfill load development before the unmined section between the panels could be extracted, thus simulating a sequential retreat strategy.

The finite element model was defined by a total of 128 eight-noded quadratic serendipity elements. The vertical boundaries of the model were established at the top of the limestone formation (49 m above potash seam) and at mid-height of the salt formation below the mining zone (20 m below potash seam). The lateral boundaries were established 192 metres, at the mid-section of the unmined block between the panels, and 384 metres into the barrier pillar. The backfilled sections were 384 metres wide, corresponding to the length of the shortwall face.

The analysis was performed to evaluate the stress and strain conditions developed along a cross-section passing through two mined shortwall panels separated by a block of unmined ore. Mine induced stress and strain conditions and backfill load development were evaluated every 360 hours, for a period of 8640 hours, to simulate long term interaction between the backfill and surrounding strata.

Figure 5 shows the stress conditions along the mine level, in the salt back, and in the limestone formation, at different time intervals, starting from the time of panel backfilling and extending to 270 days afterwards. At the seam level, and at the time of backfilling, a zero stress condition is encountered at the boundaries of the side abutments. The stress then increased to a maximum value in the abutment zones beyond which it decreased again. The peak stress reached magnitudes 1.26 times the original virgin stress level. The effect of the lateral abutment was felt in the centre of the unmined block between the shortwall panels; that is, the initial stress conditions in the block were disturbed by mining of the adjacent panels. The lateral effect of mining reached depths in the unmined potash seam of over 190 metres.

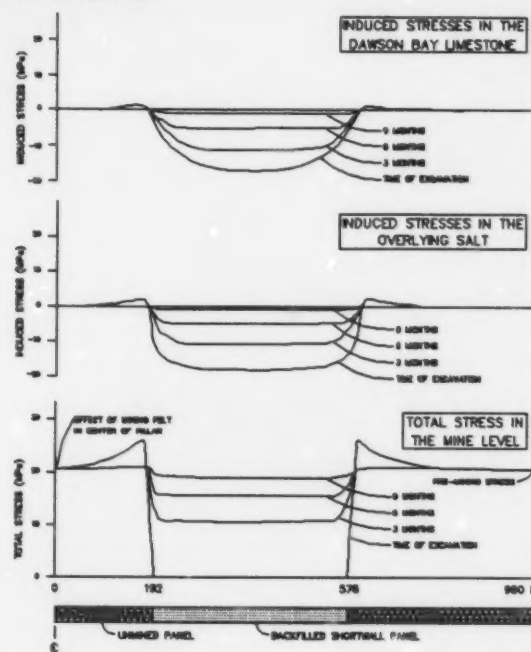


Figure 5 - Stress Profile for Two Shortwall Panels

High compressive stress zones developed in the lateral abutments extended to the overlying salt and limestone formations, but were of small magnitude and affected only a very localized region. The excess abutment stress was gradually redistributed as the backfill picked up load.

The backfilled panels did not carry any superincumbent load at placement time. Three months after placement, backfill in the mid-section of the panels carried part of the superincumbent load and provided partial support. High load carrying and support capability was developed within the backfill approximately 11 months after placement. At this point in time, mining of the ore block remaining between the backfilled panels would proceed, as the consolidated backfill would provide the necessary ground support to ensure that stable conditions are maintained during mining.

Calculated mine induced stress conditions existing in the overlying salt and limestone formations indicated that, at the time of backfilling, an arching effect, resulting from an induced zone of stress relief, had been developed surrounding each panel. The zones of decreased stress concentration encompassed each mined out panel, extending from abutment to abutment. The vertical extent of the distressed zone, at that point in time, involved the limestone formation. As the backfill developed load on a time dependent basis, a corresponding redistribution of the induced stresses took place and the influence of the induced stresses decreased to a condition of secondary importance with respect to mine stability.

Figure 6 shows the vertical strain profile occurring across the backfilled panel longitudinal section. Approximately 11% maximum strain occurred in the mid-section of each panel before the backfill

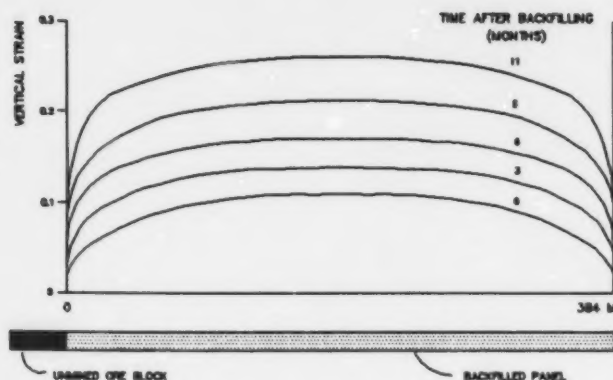


Figure 6 - Strain Profile Across a Backfilled Panel

started to develop load. The strain rate decreased with time as the backfill became stronger. At three, six, nine and eleven months after backfill placement, the mid section of the panels had reached 13.8%, 17%, 21.4% and 26% strain. The effectiveness of the backfill as a ground support mechanism is indicated by a 74% decrease in potential subsidence, achieved in the mid-section of the panels, 11 months after backfilling.

The results obtained from the analysis discussed in this section agree well with the results from the preceding analysis, and give an overall indication of the practicality of the proposed method in terms of increasing the extraction ratio without posing risk to overall mine integrity. Extensive field trials are, however, imperative for verifying the strata behaviour under actual conditions.

4 - CONCLUSIONS

Although the soft rock industry is inherently safe, due to the mining methods and extraction ratios used, the mines have a requirement to identify alternative mining methods which might provide improved resource recovery, productivity, working conditions and lower mining costs.

A salt backfill characterization study indicates that a suitable backfilling material for underground soft rock mines can be used as a support mechanism, to control mine subsidence and to allow increased extraction of ore.

Numerical studies establishing relevant geomechanical aspects resulting from the utilization of proposed methods for increasing the extraction ratio in underground soft rock operations have also been successfully designed. Shortwall retreat mining incorporating backfill and yielding pillars was indicated to be, in geotechnical terms, a favourable method for mining flat lying and continuous soft rock deposits.

Model results showed that, by using the proposed mining layouts and strategies, local and regional strata response to mining would not produce any uncontrollable situations which would pose a risk to the underground operations. The proposed method would thus provide a means for increasing the mine extraction ratio, yet preserving the integrity of the mine.

5 - REFERENCES

"VISCOT: A Two-Dimensional and Axisymmetric Nonlinear Transient Thermoviscoelastic and Thermoviscoplastic Finite-Element Code for Modelling Time-Dependent Viscous Mechanical Behaviour of a Rock Mass." Technical report prepared by Intera Environmental Consultants, Inc. to the Office of Nuclear Waste Isolation, report ONWI-437, December 1983, 226 pages.

MODELLING DAMAGE ZONE AROUND AN OPENING IN HARD ROCK
USING HOEK-BROWN INITIAL YIELD CRITERION

Khairy R. Khair and T. Chan
Applied Geoscience Branch
AECL Research
Pinawa, Manitoba
Canada R0E 1L0

ABSTRACT

It is well known and documented that excavation in deep rock can create a damage zone around the surface of the excavation. The material strength, measured in terms of Young's modulus, of this zone becomes much less than that of the surrounding medium. Estimating the thickness of the damage zone and its properties is an important factor in designing mines in rock particularly in those subjected to high in situ principal stress ratios.

This paper introduces a simple approach to estimating the change in the material properties of the rock surrounding an underground opening using a finite element technique and Hoek-Brown initial yield criteria. In this method, the first step is to solve for the stresses in a homogeneous elastic medium with constant Young's modulus. The output principal stresses are then compared with the initial Hoek-Brown yield stress value. Wherever the major stress value at an element exceeds the yield stress, the Young's modulus is reduced. Rerunning the problem with the new distribution of Young's modulus will provide another stress field. The stress comparison and reduction of Young's modulus are repeated successively until the stress state everywhere in the model lies within the yield surface.

The results have shown that significant changes occurs in the Young's modulus in the high-stress areas immediately surrounding deep underground excavation in hardrocks. The model is simple and has produced results in good qualitative agreement with other numerical models.

INTRODUCTION

Nuclear fuel waste requires safe and permanent disposal to isolate it from the biosphere. In many countries, including Canada, the most promising long-term disposal concept appears to be the use of deep geological formations that are known to have been stable for millions of years.

Currently, the Canadian Nuclear Fuel Waste Management Program is evaluating the concept of safe disposal of nuclear waste in a sealed vault constructed at a depth of 500 to 1000 m in the plutonic rock of the Canadian Shield. As part of the geoscience research program, AECL has constructed an Underground Research Laboratory (URL) in previously undisturbed granitic rock. This facility is located about 120 km northeast of Winnipeg, Manitoba.

One of the objectives of the URL is to provide in situ data to assess and improve the ability of mathematical and/or numerical models to predict the geomechanical and hydrogeological responses of the granite rock mass to excavation [1]. The Mine-by Experiment was designed to achieve this objective. This experiment, one of the Operating Phase Experiments planned for the URL [2], includes the following goals: (a) improve the fundamental understanding of rock mass behaviour and failure mechanism, and (b) evaluate the excavation damage created around the underground openings and establish the controlling factors in damage zone creation. This experiment is to be conducted at a depth of 420 m, the level where the stress situation is considered to be appropriate for the development of an excavation damage zone.

Previous experience from modelling other experiments indicated that the excavation response of the rock mass close to the tunnel wall was not well represented by linear elastic models because of the existence of a damage zone around the surface of the opening. In an effort to better represent the excavation response, Chan et al. [3] modelled this zone as an area of reduced Young's modulus. They divided the area into a number of concentric rings, each ring having a constant thickness. The value of Young's modulus of each ring was defined by a calibration technique based on the ratio of the measured to the calculated displacements. Better agreement between the measured and the predicted responses was obtained by employing this approach.

This paper introduces a model that follows the same idea, i.e., modelling the damage zone as a zone of reduced Young's modulus. The new feature in this model is that the thickness of the zone and the modulus value depend on the final stress state of each point and the type of rock around the opening.

REVIEW OF NUMERICAL MODELS FOR ROCK

Many constitutive models have been developed for geological materials. Resende [4] and Desai et al. [5] give a comprehensive review of these models and most are suitable to describe the behaviour of soft rock. A few models have been introduced to describe the behaviour of hard rock [4,6]. These models are based on a continuum representation of progressive fracturing which leads to degradation of the elastic properties. This fracturing is expressed in terms of an internal damage variable, which describes the effect of cracks on the capacity of rock to support distortional loading. An evolution law relates the damage variable [6] or the rate of damage [4] to the stress and strain history.

These models have two major drawbacks. First, the number of material constants required in the models is relatively large. In some models [6], 32 constants are needed. Although these constants can be evaluated through triaxial tests, it still requires a large number of tests. Second, the values of these parameters are path dependent, and the triaxial and hydrostatic tests only provide information for a limited number of paths.

So far these models have been successful in capturing the main features of the rock's behaviour in the laboratory. Using these models to simulate the behaviour of the rock due to underground excavation has not been completed yet.

FAILURE CRITERION

This paper introduces a model that uses the theory of elasticity in conjunction with a failure criterion to describe the degrada-

tion of the material stiffness owing to crack formation.

The basic assumption used in our model is, as stated by Goodman [7] page 74, "Normally when stresses become high enough to cause fractures in the wall of a tunnel or mine, rock simply spalls off, producing a distressed zone that drives the flow of stress away from the opening." Therefore, it is necessary to establish the critical stress value at which new fractures start to develop. This critical value has to be defined using a failure criterion. The most commonly used failure criteria for rock are the Coulomb-Mohr law, with or without tension cut-off, and the Hoek-Brown failure criterion. The one used here is based on the Hoek-Brown empirical relationship:

$$\sigma_1 = \sigma_3 + (m \cdot \sigma_1 \cdot \sigma_c + s \cdot \sigma_c^2)^{0.5} \quad (1)$$

where σ_1 is the major principal stress, σ_3 is the minor principal stress, σ_c is the uniaxial compressive strength, and m, s are material constants.

When this equation was initially developed as a failure criterion, the authors defined the failure stress as the peak stress carried by the specimens [8]. Other definitions for the failure of rock are also possible. This has been the topic of interest for many researchers including Lajtai et al. [9], who have defined the failure as the stress level at which the rock suffers permanent damage. This damage will occur at the dilation of the pre-existing axial cracks and it is the stress value at the reversal of the volumetric strain as indicated by σ_{11} in Figure 1.

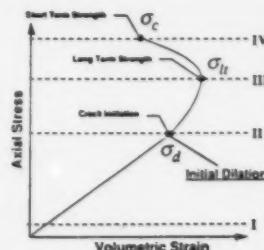


Figure 1. Volumetric strain change with axial stress for rock. σ_d mark the crack initiation and σ_{11} (volumetric strain rate reversal) mark the damage point.

Martin and Chandler [10] have considered that permanent damage will occur when the crack volume starts to grow. At this stress level new cracks start to develop as is indicated by σ_d in Figures 1 and 2. Since this value changes with the confining pressure, an envelope similar to the Hoek-Brown failure envelope could be developed using the same equation but different parameters. Figure 3 shows two Hoek-Brown failure envelopes corresponding to peak failure and new crack initiation [11]. These envelopes are produced using triaxial test data for the URL

grey granite and the crack initiation envelope is labelled initial yield criterion. It is used in our model as the critical state of stress of any element when the element starts to become damaged and its elastic modulus becomes lower.

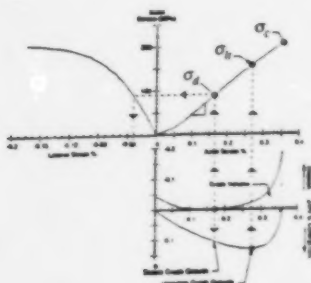


Figure 2. Crack volume growth as function of stress and strain. σ_d mark the start of crack volume increase.

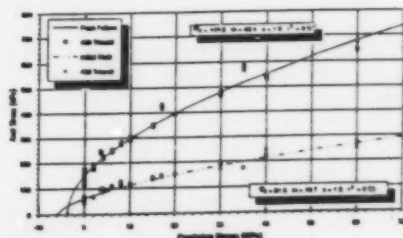


Figure 3. Hoek-Brown failure envelopes for grey granite.

MODEL DEVELOPMENT

The commercially available finite-element code ABAQUS [12] was chosen to perform the geomechanical modelling. The main features that make this code suitable for simulating the excavation include the following:

- (1) In situ stress can be incorporated into the finite-element formulation either by being applied through the input data file or through a user-supplied subroutine.
- (2) There is an element removal capability that can be used to simulate the excavation process by removing the element stiffness and the corresponding nodal loads, thus allowing the creation of a new traction-free surface.

In simulating the excavation of the tunnel, we applied an iterative method consisting of three steps:

- (1) The in situ stress is first loaded into the finite-element model in which the rock is assumed to be an isotropic, linearly elastic medium.

- (2) The elements occupying the tunnel space are removed from the mesh. The result from this step is simply the well-known elastic solution for a circular opening in an infinite medium (Kirsch solution).
- (3) The principal stresses at each element are then compared with the Hoek-Brown initial yield criterion. Wherever the major principal stress value exceeds the initial yield value, the elastic modulus of that element is reduced by 20%. As a result of this step the elastic modulus, E , of some of the elements, will be a reduced while the properties of the rest of the elements remain unchanged.

Repeating the previous three steps with the updated distribution of elastic moduli will yield a new stress field. The principal stress values are again compared with the initial yield criterion and the elastic modulus reduced wherever necessary.

This iteration is repeated until the stress state everywhere in the medium lies within the initial yield surface. The final results represent the total stress field and the total displacements in the model. The Poisson's ratio remains unchanged throughout the analysis. Figure 4 shows the flowchart of these steps.

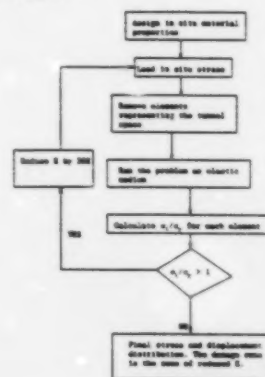


Figure 4. Flowchart of the model.

The reduction factor, 20%, is an assumption based on previous experience from modelling other experiments [9]. Reducing this factor will increase the number of iterations. It is believed that this value is appropriate for this analysis; however, a sensitivity study may be needed to quantify this parameter.

PROBLEM DEFINITION

The model described has been used to simulate the excavation of the test tunnel (Room 415) in the Mine-by Experiment. Figure 5 shows the general geometry of the Mine-by Experiment and the in situ (far field) stress [13]. This test tunnel has a circular cross section with a radius of 1.75 m.

The Mine-by Experiment is modelled here as a 2-D plane strain problem. The circular shape of Room 415 implies that only a quarter mesh is required to properly represent the excavation geometry (see Figure 6). The z-axis in the model coincides with the axis of Room 415. The x-axis and the y-axis are the axes of symmetry. A right-handed Cartesian system of coordinates is used in constructing the mesh. The x-axis is oriented parallel to the major in situ principal stress σ_1 direction, and the y-axis is oriented parallel to the minor in situ principal stress σ_3 , and positive upward. The finite element mesh has 992 four-node isoparametric 2-D elements and 1056 nodes. All the nodes are located on radial lines extending from the tunnel surface.

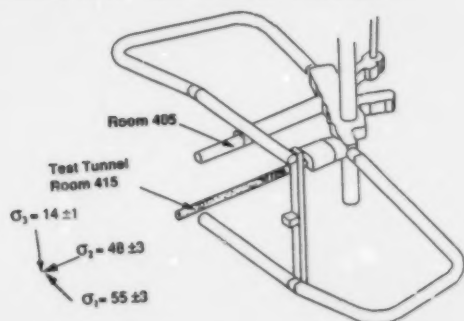


Figure 5. Isometric view of the test tunnel (Room 415).

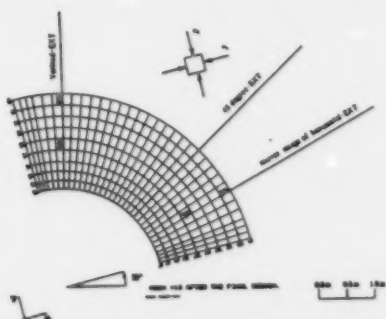


Figure 6. The finite-element mesh close to the tunnel surface.

The outer boundary of the mesh is 22 m from the centre of the room. The size of the elements is gradually increased toward the outer boundary in order to achieve optimum accuracy with a minimum number of nodes. The boundary conditions are specified as follows:

- (1) The exterior boundary is given zero displacement,
- (2) The nodes on the x-axis are given zero y-displacements, and
- (3) The nodes on the y-axis are given zero x-displacements.

It is assumed that the in situ stress field is uniform throughout the medium before the excavation. The in situ principal stress values are

$\sigma_1 = 55 \text{ MPa}$ and
 $\sigma_3 = 15 \text{ MPa}$,
 where σ_1 is oriented 15° counterclockwise from the horizontal.

The material properties employed in the geomechanical model are

In situ elastic modulus, $E = 65 \text{ GPa}$
 In situ Poisson's ratio, $\nu = 0.25$.

These values were recommended in the final design report of the Experiment [13] after they were determined through compilation of laboratory tests performed by the Canada Centre for Mineral and Energy Technology (CANMET) on rock samples from the 420 Level.

We have used the Hoek-Brown initial yield envelope as the criterion to change the elastic modulus value. Two different envelopes for two types of rocks (pink and grey granite) were considered in this study. Figure 7 shows these envelopes. The Hoek-Brown parameters of the envelopes are given in Table 1. The same Young's modulus and Poisson's ratio are used in both cases.

TABLE 1

HOEK-BROWN INITIAL YIELD ENVELOPES

Parameter	Pink granite	Grey granite
σ_c (MPa)	81.4	61.4
m	16.4	10.7
s	1.0	1.0

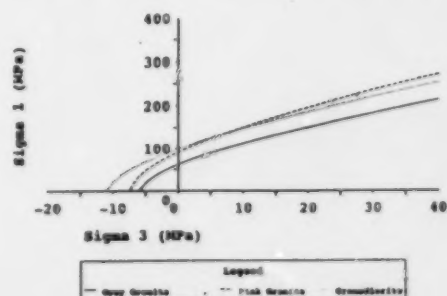


Figure 7. Hoek-Brown initial yield criteria for the URL rock.

RESULTS AND DISCUSSION

Excavation Damage Zone Thickness

The results of this analysis have led to the changing of the elastic modulus of some elements in the high-compressive stress zone.

In the case of pink granite the elastic modulus is reduced to 41.6 GPa for seven elements, and to 52.0 GPa for three elements. Otherwise the model's properties are the same as the in situ properties. Figure 8 shows the location of these elements. The thickness of this region is about 15 cm (the thickness of the elements of the first layer). This region is confined to the high-stress area ($\pm 27^\circ$ from the σ_1 direction). On the other hand, the grey granite has suffered more damage according to this model. The thickness of the damage zone is about 30 cm and extends to $\pm 45^\circ$ from the σ_1 direction. The lowest value of the elastic modulus is 8.72 GPa for five elements as shown in Figure 9.

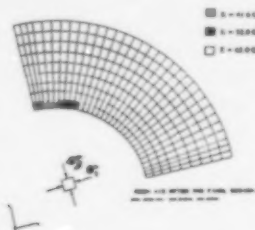


Figure 8. The excavation damage zone for pink granite.

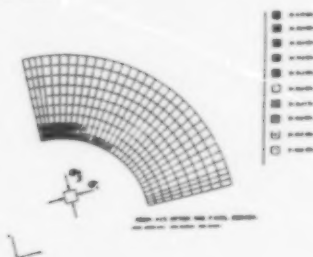


Figure 9. The excavation damage zone for grey granite.

Stress Analysis

The values of stress change are obtained by subtracting the calculated stresses, using ABAQUS, from the in situ stress values, and then converting the values from the Cartesian coordinate to the cylindrical coordinate system. Results are given for the radial and tangential stress changes about the horizontal and vertical radial lines. Figures 10 and 11 show the results for the pink granite and Figures 12 and 13 for the grey granite. It is clear that the tangential component of stress close to the tunnel surface along the vertical line has been reduced significantly in both cases compared with Kirsch's solution. On the horizontal line the rock is not damaged and still behaves like a linear elastic medium.

Comparison between the results of this model and the actual measurement made during the Mine-by Experiment will be reported in a future paper after the excavation is complet-

ed. Results of displacement are not presented here because of the space limitations; however, some of these results will be presented at the meeting.

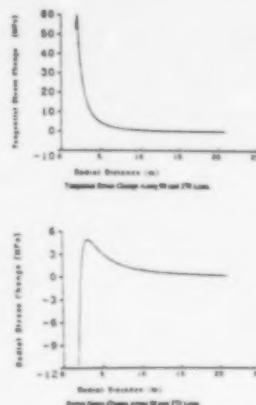


Figure 10. The tangential and radial stress distribution on the vertical line for the pink granite.

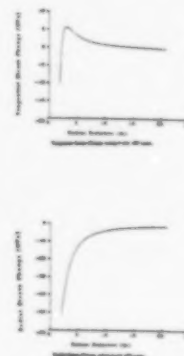


Figure 11. The tangential and radial stress distribution on the horizontal line for the pink granite.

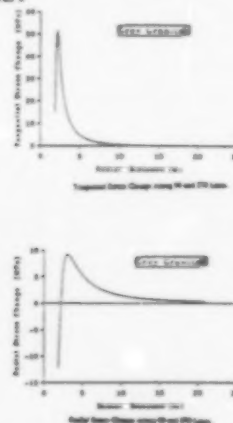


Figure 12. The tangential and radial stress distribution on the vertical line for the grey granite.

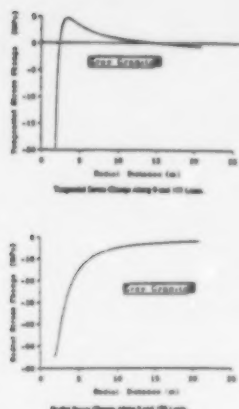


Figure 13. The tangential and radial stress distribution on the horizontal line for the grey granite.

CONCLUSION

A model using the theory of elasticity and the Hoek-Brown initial yield criterion has been developed. The model is one of three different models being used to investigate the degree of sophistication needed to predict the damage zone and the rock behaviour around the opening. The main features of this model include: (1) It does not assume specific geometry of the damage zone prior to the analysis; (2) The model is using the stress at the onset of crack growth as a failure criterion rather than the ultimate failure stress; (3) The model is simple to use. The model has shown that the geometry of the damage zone depends on the ratio of the maximum in situ principal stress to the minimum in situ principal stress, with the thickness tends to be greater in the direction of the minimum in situ principal stress.

ACKNOWLEDGEMENT

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Numerical Modelling of Backfill Failure Response Under Variable Strength Parameters

By

C.W. Palley and J.F. Archibald
Department of Mining Engineering
Queen's University at Kingston

ABSTRACT

Strength comparison studies were performed on typical cycloned tailings used as backfill with strength provided by cement addition. During this test program, cementitious agents including granulated slag and chemical additives were investigated as alternatives to Portland cement. Strength parameters were determined by subjecting backfill mixtures to both triaxial confined and direct shear failure.

These samples, cured between 14 and 224 days, demonstrated considerable variation in both long and short term strength behaviour. Tailings cemented with the granulated slag represent a viable alternative to cement-reinforced fill material if high early strength is not critical. Using Tekbent XX grout filler developed high early strength with the very rapid setting time causing significant water retention.

The structural behaviour of openings in backfill was examined using the Fast Lagrangian Analysis of Continua code. An upper stope, initially excavated in ore, was completely backfilled and a backfill opening established as drilling access to the stope below. As this stope was sequentially mined, both fill deformation and failure into the open stope were modelled. Both the degree and mechanism of failure into the open stope varied significantly with a change in the physical response characteristics. These models demonstrate the importance of accurately measuring backfill strength characteristics and the behaviour prediction possible from numerical modelling during the mine design process.

INTRODUCTION

The process of mine design has become more thorough and sophisticated with the advent of computer-aided techniques. The first aspect usually addressed through use of these techniques is the appraisal of the economic aspects of ore quality and the estimation of global and local ore reserves from the limited sample information provided from diamond drill holes. More recently efforts have turned to an assessment of geotechnical mine design parameters including the use of numerical modelling techniques. These techniques have been legislated within the Province of Ontario where the mining sequence must be modelled as a predictive measure to ensure the effective control of rock stress. This process, at the initial design stage, is often based on limited geotechnical data which is obtained on a secondary priority basis during ore sampling.

One very important aspect of geotechnical design concerns identification of parametric information on the backfill to be used during the mining process. With the advent of bulk mining techniques and single pass mining sequences, structural interaction between rock and fill within backfilled stopes comes early in the life of a mining operation. The behaviour of emplaced backfill affects the economics of an operation through its potential for dilution and hazard control. Backfill failure can result in high dilution and loss in effectiveness of controlling pillar stability.

A project to examine the technical feasibility of utilising openings in hydraulic backfill for stope access was carried out at Queen's University through funding provided by the Mining Research Directorate of Sudbury, Ontario¹. In conjunction with other backfill research being funded by NSERC², it was decided to initiate a complete laboratory analysis of the physical and engineering properties of a typical hydraulic fill material. The purposes of this study are as follows:

1. to permit comparison of strength results for tailings mixtures manufactured using different cementing agent materials, and
2. to obtain backfill physical parameter information suitable as input data for numerical modelling procedures.

The cementing agents selected for evaluation included Normal Portland cement, Reiss Lime slag, Fosco Tekbent grout and a polyurethane resin gel, produced by MSE Multiurethanes Limited.

This paper describes the laboratory tests which were performed and the results obtained. It is hoped that this work may be of help to operations designing a testing program to obtain fill data or wishing to use realistic backfill parametric data, where none may be presently available, for the purpose of initial mine design. In addition, the results of two numerical models using differing backfill input parameters demonstrate that backfill stability analysis can be predicted through the use of such technology.

It should be noted that this paper deals exclusively with characterization and modelling of hydraulic backfill only. Whereas cemented rock fills are important in certain bulk mining situations, their laboratory and in-situ characterization presents significant difficulty. Numerical modelling of rock fills is complicated by this lack of post placement data.

The backfill material used in this analysis was provided by Denison Mines Limited, one of the sponsoring companies of the Mining Research Directorate project.

TAILINGS CHARACTERIZATION

The tailings material, an unclassified mill product of the Denison Mines Limited operation at Elliot Lake, Ontario, was subjected to a standard series of laboratory physical property evaluation tests¹. Such as-received materials were subjected to examination and characterization of placed bulk density (in loose and consolidated states, and under varying moisture contents), true solids density, porosity, permeation rate, size distribution and direct shear strength behaviour. Data was determined to provide base estimates of its suitability as fill material when utilized in a non-cemented fashion. The size distribution data is presented in Figure 1 and the physical property data for as-received, non-cemented tailings is presented in Table 1.

TABLE 1 - Physical Characteristics of As-Received and Tested Tailings

<u>Density Parameters</u>			
Dry, loose poured bulk density, γ (g/cm ³) -	1.47		
Dry, consolidated bulk density, γ_c (g/cm ³) -	1.96	$(\sigma_{consolidation} = 664 \text{ kPa})$	
Dry, true solids density, ρ (g/cm ³) -	2.77		
Dry, loose poured porosity, n -	46.9%		
Moist, consolidated bulk density, γ'_c (g/cm ³) -	2.23	$(\sigma_{consolidation} = 664 \text{ kPa})$	
(@ 14.1 % moisture content)			
<u>Long Term Percolation Rate (24 hour stand time) -</u>		14.6 cm/h	

<u>Non-Cemented Tailings Material Direct Shear Parameters</u>			
	<u>Cohesive Strength, C (kPa)</u>	<u>Internal Friction Angle, ϕ (°)</u>	
Dry tailings	33.7	28.0	
Moist Tailings	14.9	29.4	
(@ 14.1 % moisture)			

<u>Particle Size Size Distribution</u>			
<u>Mesh Size</u>	<u>Opening Size (mm)</u>	<u>Cumulative % (by weight) Passing</u>	
8	2500	97.9	
14	1180	95.0	
20	850	89.3	
25	625	81.2	
40	375	65.3	
60	250	43.3	
100	150	24.5	
150	100	12.9	
200	75	6.4	
250	50	3.4	
325	40	2.3	

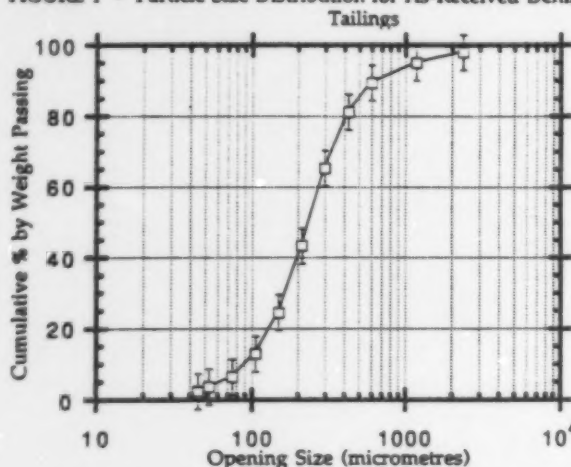
The tailings product demonstrated a restricted fines content (only 10% finer than 170 mesh and 2% finer than 325 mesh), which was also determined to exhibit a Coefficient of Uniformity (C_u) value approximating 2.90. Materials which exhibit C_u properties approximating 5.0 display dense packing states, and, consequently, good uncemented strength. Where either very low or very high C_u conditions are exhibited, aggregate materials often display low density and poor strength characteristics.

The tailings product was therefore characterized as a medium-to-coarse, well graded aggregate material which would exhibit very suitable permeation rate behaviour (14.6 cm/h) when placed as a non-cemented slurry. In the mining industry, where permeation rates of 10.0 cm/h can be exhibited by placed slurry fill products, satisfactory drainage capability, low liquefaction potential and early access to fill working floors result.

Non-cemented tailings material was also subjected

to bulk shear failure to determine non-cemented material strength parameters. All samples were tested under fixed rates of axial strain and at two arbitrary moisture content states (dry and at approximately 14% moisture) to evaluate variation in non-cemented material strength behaviour normally developed over the range of tailings moisture states encountered in-situ following slurry placement. In all tests, samples were subjected to pre-consolidation loading (equivalent 50 metre fill head loads) in order to duplicate expected field density conditions.

FIGURE 1 - Particle Size Distribution for As-Received Denison



Bulk shear examination indicated that moderate frictional and low level cohesive strength behaviour could be developed when the material was pre-consolidated. Cohesive strengths, varying between 17 - 54 kPa, and uniform internal friction angle values (at approximately 28.5°) were indicated for the material.

BACKFILL PREPARATION and STRENGTH EVALUATION PROCEDURES

Upon completion of initial characterization work, tailings samples were consolidated using normal Portland cement and other commercially-available consolidating agents to determine mixture strength behaviour and provide a comparison between various agents. Such information is necessary to provide a measure of the physical property variation which may be exhibited and to provide quantitative data from which engineering projections concerning fill suitability may be made.

The primary incentive to replace Portland cement in consolidated fill is its high cost. Where alternate materials can be used to produce equivalent or better fill strength behaviour, at lower cost, their application within the mining industry would be welcomed. Alternate consolidation agents which were selected for review in this project included:

a) Reiss Lime - ground, zinc-based Algoma smelter slag, known to exhibit pozzolanic behaviour, which is used within the mining industry as a common and cheaper replacement for Normal Portland cement in the manufacture of consolidated backfill. This product is produced by the Reiss Lime Company of Blind River, Ontario.

b) Tekbent IX Grout² - a single component, finely ground, dry chemical grout material produced by Fosco Canada Ltd., Guelph, Ontario.

c) polyurethane grouts³ - such grouts consist of single-component polyurethane liquids, mixed with catalyzing agents to alter reaction set times, which are designed to react with water to produce an expanding foam material. When added to backfill slurry, encapsulation of fill aggregate materials within the resulting foam matrix produces rapid fill "set" during placement. Excess slurry water is chemically bound within the fill mass and is not lost by percolation

through the mass. Polyurethane grout materials tested (MME Ultrafine and MME Universal polyurethane resins, each consisting of a prepolymer and alkylamine accelerator) are produced and sold by MME Multiurethanes of Mississauga, Ontario.

Dry mixtures of as-received tailings and the various consolidating agents, at various ratios by weight, were prepared. Bulk quantities of tailings/cement or tailings/grout, at two arbitrary ratio combinations corresponding to 30:1 and 20:1 tailings/cement, were fabricated. Materials were manufactured within the laboratory and tested under the following conditions:

- all samples manufactured at 70% pulp density
- curing humidity maintained at 100%
- curing intervals maintained between 14-112 days for Portland cement-consolidated samples and 24-224 days for slag-consolidated materials
- samples tested were manufactured as 5 cm x 10 cm undrained cylinders
- samples were subjected to triaxial (confined) compressive strength failure; in selected cases, direct shear failure was also conducted.

Mixtures were prepared using both potable and mine water to determine whether mine water-borne impurities might affect fill strength development. Mine, rather than potable, water is utilized at most operations to manufacture backfill and typically contains hydrocarbon contamination or may exhibit considerable pH variation from site to site.

Backfill mixtures using normal Portland cement and Reiss Lime slag, were cast and permitted to cure both in the form of 5 cm diameter by 10 cm long, cylindrical specimens for triaxial failure evaluation and within 15 cm diameter by 18 cm deep laboratory shear cells for direct shear failure evaluation. All specimens prepared using grout consolidation agents, were cast within shear cells only. Due to the rapid set nature of all of the grout agent materials utilized, with reaction times measured in terms of minutes or fractions of minutes, triaxial sample preparation using grout cementing agents was not possible.

Due to the need to prepare and mix large quantities of fill material for each direct shear test specimen, and to limited tailings availability, only four samples were tested. Accordingly, direct shear sample tests were conducted only for short term cure intervals of 14 and 28 days.

STRENGTH RESULTS

A summary of strength and physical property measurements observed during these tests is illustrated in Table 2(a), for Portland cement and Reiss Lime cured specimens, and Table 2(b), for grout agent cured specimens.

Graphs illustrating variation in strength behaviour versus both time of cure and consolidation material type are shown in Figures 2, 3, 4 and 5.

Figure 6 represents a comparison of unconfined compressive strength behaviour which was demonstrated by Portland cement and Reiss Lime consolidated backfill mixtures after 28 and 112 days of cure, respectively.

Figure 7 is used to illustrate measured 28 day direct shear strength parameters derived for mixtures containing each of the five different consolidation agents utilized.

Portland Cement-Reinforced Fills

All Portland cement-reinforced fill mixtures were demonstrated to develop considerable unconfined compressive strength improvement over initial cure intervals approximating 112 days, as illustrated in the data of Table 2(a) and Figure 2. During this period, minimal strength gains were observed for all mixtures tested in the first 28 days of cure, with significant strength increases being shown to develop only between the 28 - 112 day cure stage. Between 56 and 112 days of cure, the trend of increasing strength was demonstrated to gradually level off, with no peak

TABLE 2 (a) - Summary of Denison Backfill Strength Test Results for Normal Portland Cement and Reiss Lime Slag Consolidated Mixtures

Cement Type	Mix Ratio (water type)	Cure Age (days)	TRIAXIAL TESTS			DIRECT SHEAR			Density (g/cm ³)	% Water
			UCS (kPa)	Cohesion (kPa)	Friction Angle (degrees)	UCS (kPa)	Cohesion (kPa)	Friction Angle (degrees)		
Portland	30/1 (mine)	14	146	41.3	31				1.83	15.8
	30/1 (potable)	14	140	35.9	33.7	247	46	34.4	1.86	16
	20/1 (mine)	14	250	66.4	35				1.91	14.9
	20/1 (potable)	14	254	72	20.9	370	96	38.6	1.92	15.3
Portland	30/1 (mine)	28	167	45.9	32.4				1.83	13.7
	30/1 (potable)	28	139	39.9	30.3	38.4	9	39.8	1.82	14.7
	20/1 (mine)	28	252	71.3	31				1.85	15.3
	20/1 (potable)	28	255	57.8	31.4	192.1	47.1	37.8	1.91	14.9
Portland	30/1 (mine)	56	239	66.4	41.3				1.81	12
	30/1 (potable)	56	266	71.4	33.4				1.8	12.6
	20/1 (mine)	56	515	120.2	40.1				1.88	14.7
	20/1 (potable)	56	527	136.4	38.7				1.87	13.1
Portland	30/1 (mine)	112	312	75.6	38.3				1.9	14.3
	30/1 (potable)	112	310	77.7	36.7				1.87	13.6
	20/1 (mine)	112	736	175.1	38.5				1.95	11.7
	20/1 (potable)	112	704	167.4	39.1				1.95	13
Reiss Lime	30/1 (mine)	28	115	25.2	42.5	259	66.8	38.3	2.08	16.1
	30/1 (potable)	28	226	65.9	31.6	130	32.4	38.1	2.04	16.1
	20/1 (mine)	28	152	36.4	34.5	0	0	35.9	2.02	14.4
	20/1 (potable)	28	146	35.2	38.5	181	44.5	37.6	2.09	14.9
Reiss Lime	30/1 (mine)	112	802	112.4	45.7				2.02	14.2
	30/1 (potable)	112	752	155.9	46				1.96	13.3
	20/1 (mine)	112	1074	218.8	43.3				2.03	14.4
	20/1 (potable)	112	988	201.9	45.6				2.03	13.9
Reiss Lime	30/1 (mine)	224	911	195.5	39				1.92	7.5
	30/1 (potable)	224	921	223.1	38.8				1.94	9.8
	20/1 (mine)	224	1340	297.7	42.1				1.96	10.2
	20/1 (potable)	224	1085	227.4	44.5				1.95	10.4

strength behaviour being developed before completion of the initial 112 day cure interval.

Little strength difference was observed between backfill mixes manufactured at similar cement concentrations using either potable or mine water. In view of this observation, no benefit was shown for fabrication of fill using potable water in place of mine water when normal Portland cement was used as the fill consolidation agent.

TABLE 2 (b) - Summary of Denison Backfill Shear Strength Test Results for Grout Agent Consolidated Mixtures

Results for Denison Tailings Plus Grout Agent Additions			DIRECT SHEAR STRENGTH RESULTS		
Grout Agent Used	Mixture Components and Combination Ratio		UCS (kPa)	Cohesion (kPa)	Friction Angle (degrees)
Portland cement with Pseudo Tailings XI	20/1 tails/cement dry weight mixture plus 5% by weight mix Tailings XI grout solids (ie - 13/1 @ 28 day cure)		600	232	14.6
Portland cement with Pseudo Tailings XX	40/1 tails/cement dry wt. mixture plus 1.5% by weight mix Tailings XX grout solids (ie - 20/1 @ 28 day cure)		227	66.7	29.2
Portland cement with MME Universal gel resin	20/1 tails/cement dry weight mixture plus 5% by weight mix gel resin (ie - 13/1 mix @ 28 day cure)		241	88	17.7
MME Ultrafine gel resin	20/1 tails/gel resin weight mixture		0	0	46.3

TABLE 3 - Model Input Parameters

In-Situ Boundary Stress Conditions			
Depth of Burial -	1000 metres		
Vertical Stress Gradient -	0.027 MPa/metre		
Horizontal Stress Gradient -	1.8 x Vertical		
Elastic Parameters			
Material	Young's Modulus	Poisson's ratio	Bulk Density
Ore	40,000 MPa	0.18	3300 kg/m ³
Waste	34,000 MPa	0.20	2700 kg/m ³
Backfill	380 MPa	0.35	2000 kg/m ³

Backfill Strength Parameters

Fill Type	Friction Angle	Strength (kPa)	
		Cohesive	Tensile
20/1 Tails/Portland @ 112 Day Cure	29°	175	70
20/1 Tails/Portland/Tailings XX @ 112 Day Cure	19°	232	70

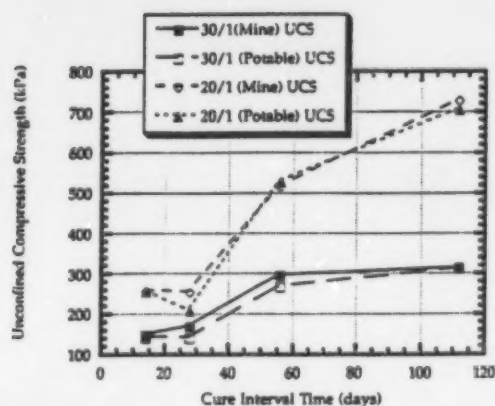


FIGURE 2 - Unconfined Compressive Strength Results for Denison Tails/Portland Cement Binder Mixtures

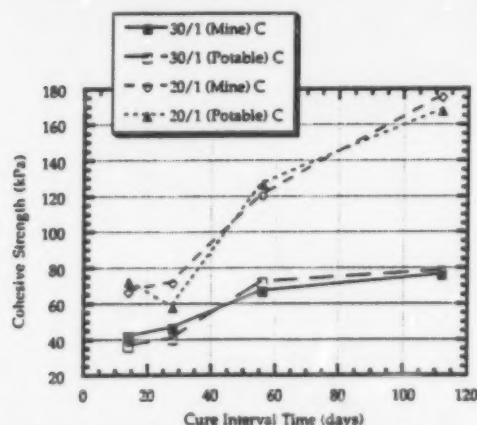


FIGURE 3 - Cohesive Strength Results for Denison Tails/Portland Cement Binder Mixtures

Reiss Lime-Reinforced Fills

All Reiss Lime reinforced fill mixtures were demonstrated to develop maximum unconfined compressive strengths at cure intervals exceeding 224 days, with significant and rapid increases occurring during the initial 112 days of cure, as illustrated in the data of Table 2(a) and Figure 4. Between 112 and 224 days of cure, the trend of rapid strength increase was shown to gradually decrease, though not to stabilize, for any of the mix ratios or water types utilized.

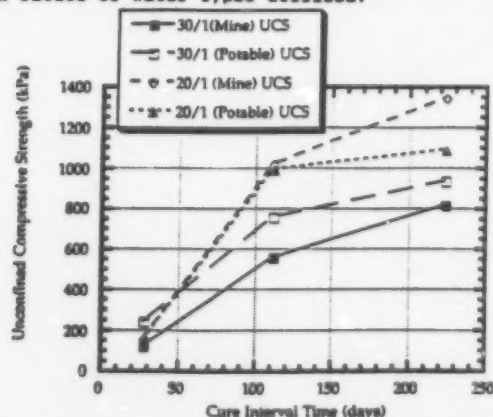


FIGURE 4 - Unconfined Compressive Strength Results for Denison Tails/Reiss Lime Binder Mixtures

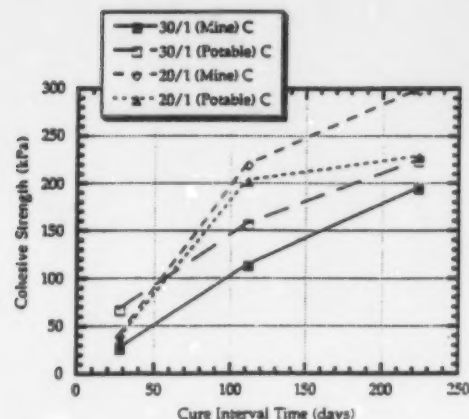


FIGURE 5 - Cohesive Strength Results for Denison Tails/Reiss Lime Binder Mixtures

All Reiss Lime fill products, while requiring greater time to develop peak strength behaviour relative to Portland cement fill products, were demonstrated to develop greater ultimate strengths for equivalent tailings to binder mix ratios (Figure 6). Similarly, Reiss Lime fabricated fills showed consistently higher friction angle behaviour than did Portland-manufactured products at similar cure ages.

In contrast to results observed for fills manufactured with Portland cement, Reiss Lime mixes evidenced more noticeable strength variability dependent upon the type of water which was used during fill mixture fabrication.

Grout-consolidated Fills

Two types of grout-consolidation agents were utilized in the fill manufacture process and were subjected to strength examination using direct shear techniques following 28 days of cure.

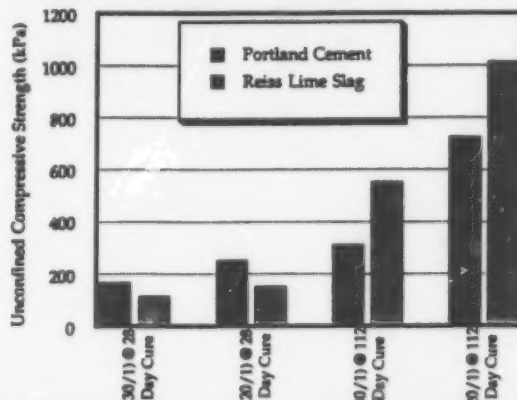


FIGURE 6 - Unconfined Compressive Strength Results for Portland- and Reiss Lime-Consolidated Fill Mixtures after 28 and 112 Days of Cure (prepared using mine water only)

The Tekbent XX grout material is available as a finely-ground powder and was dry mixed with tailings and other cement agents prior to preparation of backfill slurry batches; the MKE Universal Gel resin is available only in liquid form and was mixed with other fill components. In mine fill applications, both of these agents would be introduced into the slurry feed at the point of line discharge into stopes due to their short reaction set times.

The Fosco Tekbent XX material, which claims to rapidly lock up molecular water when mixed in the form of a slurry, has been observed ⁷ to produce rapid

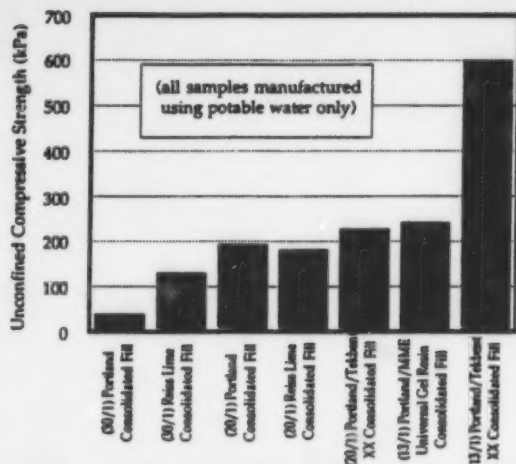


FIGURE 7 - Direct Shear Strength Results for Consolidated Fill Mixtures. Tested after 28 Days of Cure (prepared using potable water only)

product set within minutes following both slurry mixing and backfill material casting within shear cells. Such rapid set capability resulted in little or no seepage water escaping from the casting cylinders. Such complete utilization of the aqueous fraction of the backfill slurry may prove beneficial in reducing pumping requirements but may also preclude effective fill placement due to limited flow time of solids from the slurry line to stope boundaries.

Summary shear strength data illustrated in Table 2(b) and Figure 7, for all consolidation agents reviewed in this study, can be used to directly compare the effectiveness of the various agents in tailings reinforcement.

Shear failure analysis on Portland and Reiss Lime reinforced fill materials indicated that beneficiation of frictional resistance to shear was developed with increased cure. The range of friction angle response for Portland consolidated fills at 112 day cure ranged between 37° - 39°, up from values ranging between 30° - 32° after 28 days of cure. Similar friction angle response varied between 43° - 46° at 112 day cure and 36.5° - 42.5° at 28 day cure for the Reiss Lime consolidated fills. For each of these cases of tailings consolidation, friction angle response showed improvement over values which were observed for the non-cemented, as-received tailings, where values ranged between 28° - 29.4°.

Where grout and resin consolidated materials were utilized to consolidate tailings, a noticeable drop in friction angle shear response was observed. When Tekbent XX grout was mixed in equal weight proportion with Portland cement to reinforce tailings, friction angle values were observed to decrease with increasing grout percentage additions. Approximate friction angle values of 29.2° and 14.6° were derived for 20:1 and 13:1 mix ratios respectively. In the singular case where Universal Gel resin was added, 28 day friction angle response was observed to be 17.7° or approximately half that which was exhibited by Portland or Reiss Lime consolidated mixtures for a similar cure age material.

NUMERICAL MODELLING OF FILL PERFORMANCE

As part of the process of evaluating the stability of openings in backfill, a numerical model was developed using the FLAC code, which is marketed by Itasca Limited of Minneapolis, Minnesota⁷. A variety of backfill materials, under varying stope geometry and material physical property conditions, have been physically modelled⁸ to failure to provide a measure of stability estimation for both cut-and-fill and bulk mining operations. Where accurate numerical modelling procedures can be designed to generate similar results, considerable savings in both costs and time to generate and review fill failure data may result.

The FLAC model which was adopted was used to simulate a mining zone which is 60 metres wide by 120 metres high, with a 10 metre ore zone dipping vertically through the center of the model, as shown in Figure 8. The ore and host rock were modelled as elastic materials and the backfill as a Mohr-Coulomb plastic material with input data and failure criteria as described by the FLAC operating manual(6). The model was used to simulate two, 50 metre high stopes which were being mined downwards, with the upper stope being completely excavated first, then filled. The drill drift for the lower stope was excavated through the backfill, which occupied the upper stope, and the lower stope was later excavated. Lower stope excavation was accomplished by the removal of three, 15 metre lifts above the initial 5 metre undercut. The excavation process is outlined in Figure 8, and the empirical modelling parameters which were used are listed in Table 3. The assumed geometry is typical of many bulk mining situations, the only difference being that the drill opening is established in the backfill as opposed to being in the rock within the sill pillar below.

In order to compare backfill behaviour in the model, all parameters were kept constant with the exception of the backfill friction angle and cohesive strength values. Both model runs terminated when failure of the backfill into the lower open stope took place, the first occurring at modelling step 8340 and the second at step 8260.

The displacement behaviour of two backfill materials; a 20:1 Portland cement reinforced fill and a 20:1 Portland cement/Tekbent grout reinforced fill, and conditions of opening stability were shown to be quite different. Figure 9 illustrates the modelled backfill behaviour of both materials which existed at a time just prior to excavation of the final lift within the lower stope. The opening within the 20:1 Portland cement reinforced fill was shown to be stable, with only slight relaxation of the opening roof occurring. A similar opening, modelled to exist within the 20:1 Portland and Tekbent XX grout reinforced fill, showed extensive convergence and almost complete closure was predicted.

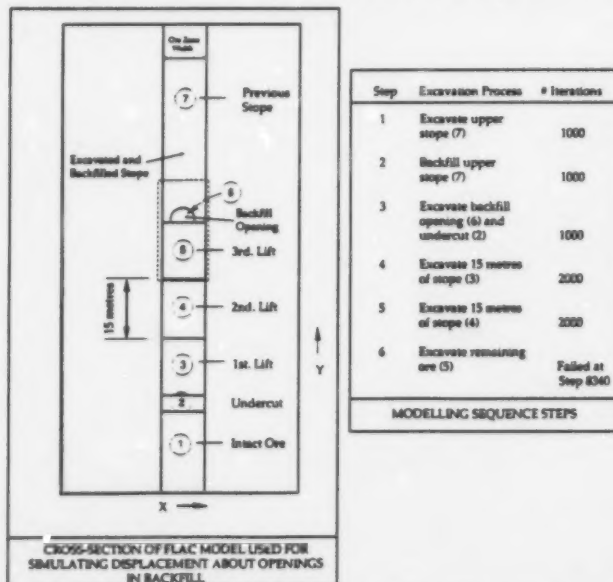


FIGURE 8 - Illustration of the Physical Stope Model and Sequence of Extraction Leading to Backfill Opening Failure

Figure 10 illustrates the conditions of backfill displacement at the time of respective model failure into the lower stope after final sill extraction. The Portland cement reinforced backfill was shown to fail locally about the initially-created fill opening and to create a stable arch at a level approximately 8 metres above the base of this opening. The Portland cement/Tekbent XX grout reinforced backfill product was shown to undergo considerably more extensive failure,

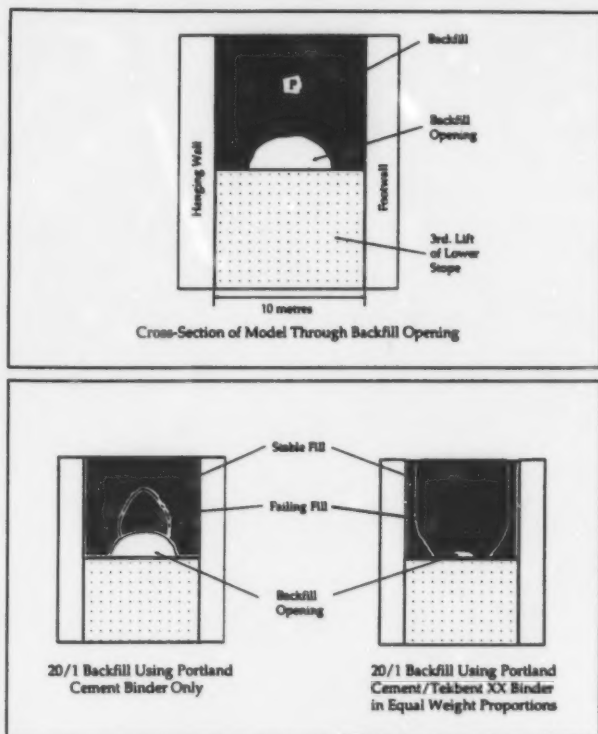


FIGURE 9 - Illustration of Stable and Deformed Backfill Zones About Fill Excavation Prior to Removal of 3rd. Lift, Lower Slope

resulting in funnel flow of backfill along the entire length of the walls of the upper slope.

These conditions are also demonstrated in Figure 11, where plots of the vertical, downward displacement, (Y), of a point originally lying 2 metres above the center of the back of the fill opening are shown. In the case of the Portland reinforced fill mass, some minor relaxation occurred within the fill as the backfill opening was initially excavated. The drift remained stable until final excavation of the sill pillar below the opening. In the case of the Portland cement/Tekbent XX reinforced fill mass, closure of the opening was shown to be substantial immediately following its excavation within the fill. As mining in the lower slope progressed, the rate of closure slowed somewhat but continued until the point at which final sill excavation took place, whereupon catastrophic fill failure occurred.

The results of these models demonstrate the following:

- 1) that complete backfill characterization is essential if one is going to accurately attempt to predict backfill failure. In the case of the Portland cement/Tekbent XX reinforced backfill material, for example, if one were to calculate a free standing fill face limit based solely on normal parametric strength data, unconfined compressive strength (UCS), cohesive strength (C), and internal friction angle, and assuming planar shear failure conditions only, serious errors could result. For those materials exhibiting very low friction angle behaviour, the failure response of this particular backfill mixture, the potential mode of failure becomes one of cohesionless slump upon initial displacement. In a situation where the backfill material may not be exposed along free faces, where high early strength may be beneficial and where reduction in decant requirements during placement is desired (in filling a failed orepass, for example), the use of Tekbent XX additives would be highly recommended. As well,
- 2) the use of a numerical modelling code, such as FLAC, permits one to make comparisons between physical responses of various backfill materials. If backfill is

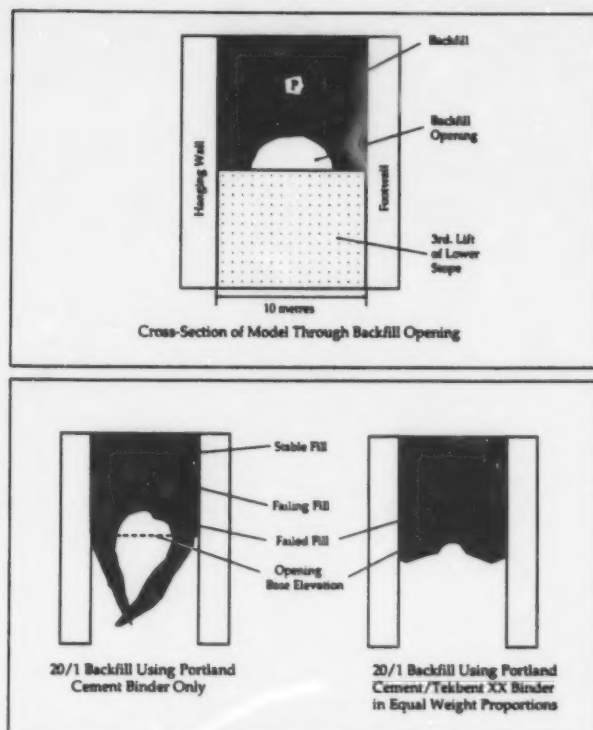


FIGURE 10 - Illustration of Stable and Deformed Backfill Zones About Fill Excavation After Removal of 3rd. Lift, Lower Slope

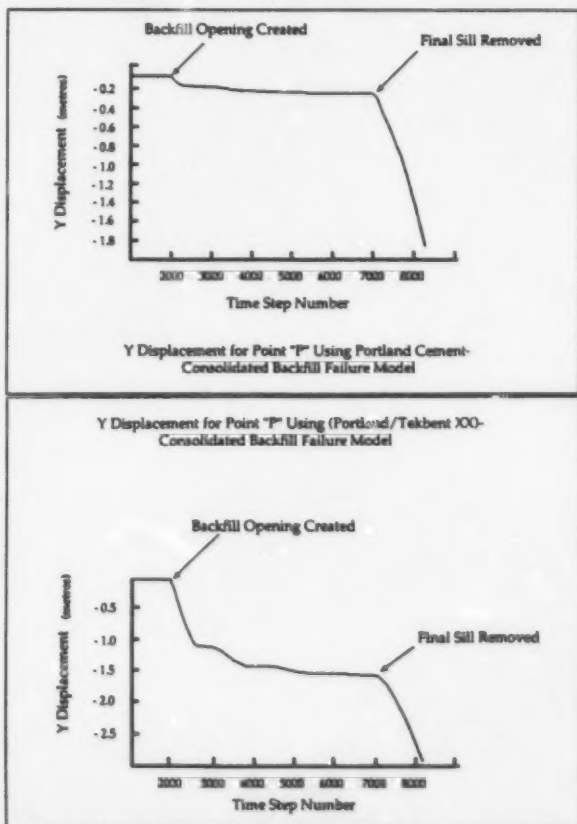


FIGURE 11 - (Y) Displacement Plots for Backfill Modelled Response Over Excavated Opening

to be undermined, as was done in this modelling case, one can obtain estimates of the amount of dilution which can be expected. By examining the response of various fill types, one can better predict the suitable consolidation agent ratio and type of consolidation agent to use in achieving the desired engineering result. It must be remembered that such fill physical property characteristics and excavation responses provide only relative comparisons, and that absolute data concerning such capabilities can only be determined from field measurements. The use of modelling, however, should give rise to more logical first applications.

CONCLUSIONS and RECOMMENDATIONS

At similar 28 day cure stages, no significant unconfined compressive strength variation was shown between backfill materials manufactured at a 20:1 tailings:consolidation agent ratio for Portland cement, or a 50/50 mixture of Portland cement/Tekbent XX grout agents. For the Reiss Lime consolidating agent, lower early strength characteristics were observed.

Where richer 13:1 mixtures of Portland cement/Tekbent XX fill were manufactured, significant early shear strength improvement was shown over all of the fill materials cast at a 20:1 mix ratio. In contrast, no fill strength benefits over leaner 20:1 mixes was demonstrated by the combination of Portland cement with MME Universal Gel resin to create a 13:1 mixture of Portland cement/Universal Gel fill material.

What was shown to be significant in such interval results was the means by which backfill strength mobilization occurs between samples. In all mixtures utilizing either Tekbent XX grout or the Universal Gel resin, strength improvement was demonstrated to derive from increased cohesion bonding, with a consequent loss in frictional strength behaviour, even relative to non-cemented tailings material. For both mixtures manufactured using only Portland cement or Reiss Lime agent additions, fill strength was demonstrated to derive from improved frictional resistance to failure, with only gradual improvement in cohesion bonding strength.

Due to limited availability of tailings, only Portland and Reiss Lime reinforced fill materials were manufactured in sufficient quantity to permit long term cure strength observation. After periods of cure up to 112 days it was observed that Reiss Lime reinforced backfill materials demonstrated higher strengths than did normal Portland cement reinforced fill materials. In the longer term, it would be expected that similar ultimate strength relationships would also occur.

The choice of fill consolidation agent material must be determined according to constraints in terms of raw material cost factors, engineering design considerations and acquisition limitations. No cost comparison was performed between the agent materials utilized for tailings consolidation. Due to the specialty nature for fabrication of each of the grout materials tested, it may be anticipated that adoption of such materials in place of either Portland cement or Reiss Lime would result in higher costs to mine operators.

Both Portland and Reiss Lime materials were shown to produce equivalent fill strength behaviour relative to specialty grout/gel products, and each could be used in place of the other to satisfy engineering requirements for in-situ stope support. The Reiss Lime product is produced at only one site in Ontario, although other sources of pozzolanic slag materials may be available as results do demonstrate that pozzolanic slag products can act as effective binders.

In field use, both Tekbent XX grout and Multiurethane gel agents, due to their short reaction times, must be mixed with slurried tailings at or close to the point of discharge from fill lines into working stopes. The cost to procure such agents, anticipated problems in handling/mixing materials on site and insignificant strength benefits, relative to existing consolidation agent materials, would hinder implementation of these agents for backfill consolidation purposes.

The results of two numerical models have demonstrated the requirement to provide complete characterization of fill strength properties where alternate fill binder agent use may be anticipated. These models also demonstrate the effectiveness of numerical models for providing additional information to the decision maker during early design process steps.

ACKNOWLEDGEMENTS

Financial support from the Natural Sciences and Engineering Research Council (NSERC) and from the Mining Research Directorate (MRD) is greatly appreciated. The information presented in this publication encompasses a broad spectrum of research and investigation which has been actively pursued for a period of years and which will continue in the future. Our appreciation is extended to the funding bodies, and particularly to Denison Mines Limited, for their considerable support.

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AN EXPERIMENTAL AND NUMERICAL SIMULATION OF THE POST PILLAR CUT AND FILL MINING METHOD

M. J. Forster, J. F. Archibald, W. F. Bawden, A. J. Hyett,

Dept. of Mining Engineering, Queen's University
Kingston, Ontario

ABSTRACT

Scale modelling on intact rock cylinders surrounded by backfill was used to simulate post pillars. Finite difference modelling was also used to simulate post pillar mining. The results from both studies show qualitative agreement, and provide an interpretation of the mechanisms of post pillar yielding and the effect of fill support.

INTRODUCTION

In the post pillar cut and fill mining technique, slender pillars are left to provide permanent support. A typical post pillar pattern is shown in Fig. 1. As mining continues upwards, tall pillars are left surrounded by cemented backfill. Within the mine, only the top of the pillars can be seen above the backfill. Typical pillars have plan dimensions of 5 by 5 meters, and are located on a regular pattern such as 11 m spans between the pillars.

In deep ore bodies post pillars are not sufficiently large to support induced loads and are, in fact, designed to fail gradually. The pillars yield soon after they have been excavated, but continue to provide support at a reduced level. Backfill supports the yielded pillars, and prevents additional pillar deterioration, hence maintaining some degree of support to the excavation span.

This study was based on post pillar mining operations carried out in the Sudbury area, at depths of about 1700 m. Physical modelling on intact rock cylinders surrounded by backfill was used to scale model post pillars. Finite difference modelling was also used to simulate the post pillar interactions with the surrounding rock mass. The rock type in this study is a granite breccia, while the ore is a granite breccia containing sulphide stringers.

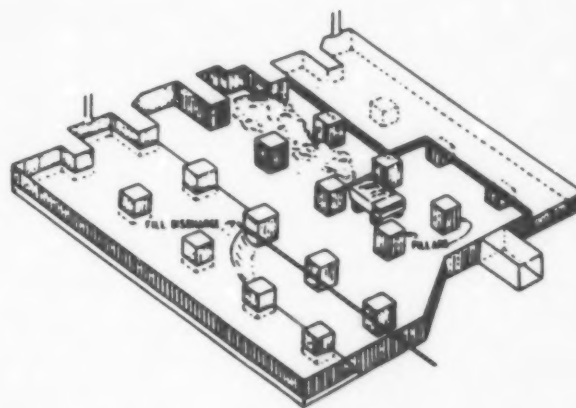


Fig. 1 Post Pillar Mining (after Barrett et al 1980).

SCALE MODELLING

The rock samples for the physical modelling tests were selected from core specimens taken from pillar blocks within Falconbridge's Strathcona Mine. Archibald (1976) used a servo-controlled stiff testing machine to generate complete specimen failure, providing results that included the post yield ranges.

Sequential Failure of Fill Constrained Samples

To include the effect of fill confinement, and the increase in pillar height with mining, a testing procedure was adapted to model a typical pillar excavation sequence, illustrated in Fig. 2. The laboratory simulation of this mechanism is shown in Fig. 3.

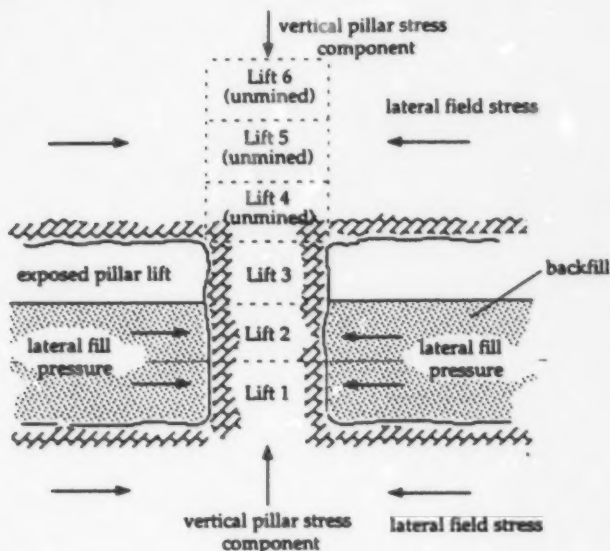


Fig. 2 Sequence of Post Pillar Development.

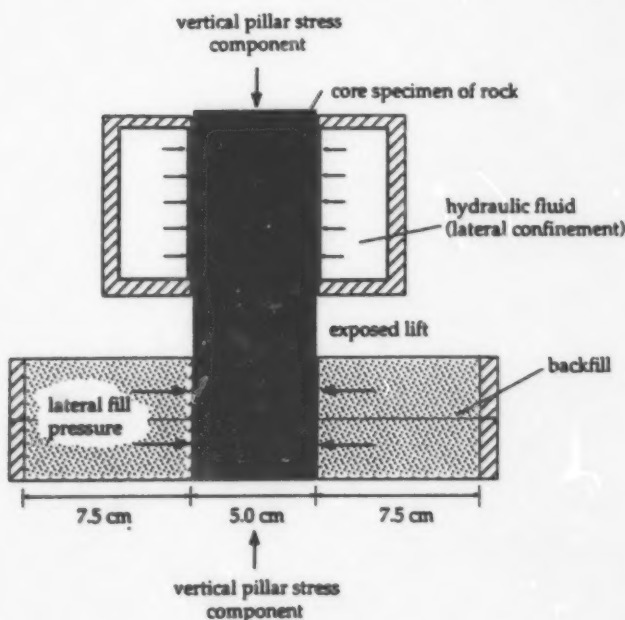


Fig. 3 Post Pillar Model Simulation.

Tailings fill and cement as a slurry were placed in confining aluminum rings and subjected to confining stress, to produce cemented fill that applied lateral pressures consistent with measured *in situ* conditions. The core segment above the exposed interval was hydraulically confined to simulate the *in situ* lateral stress field above the exposed post pillar.

Axial sample stress was applied until each sample showed evidence of existing in a post-yield failure state. To simulate the vertical progression of mining the excavated pillar zone was lengthened and further confined by fill. An extraction "lift" is defined as the excavation of a successive mining working level. The triaxial jacket above the current lift was shortened sequentially to simulate the excavation of additional unfractured lifts.

Load Deformation Behaviour

The initial lift failed by spalling and formation of slabs along the core surface. All successive lifts exhibited failure corresponding to a post yield state. As the number of pillar lifts increased the specimen load carrying capacity was seen to decrease.

A typical load deformation sequence is shown in Fig. 4, with a backfill prepared at 20:1 (tailings:cement by dry weight). The core specimens that represented tall post pillars had a post peak strength approximating 20% of the unconfined compressive strength of short pillars.

The load deformation results, correlated with sections cut from the samples, showing that progressive failure continued to occur in the lower, initial lift zones as pillar height increased.

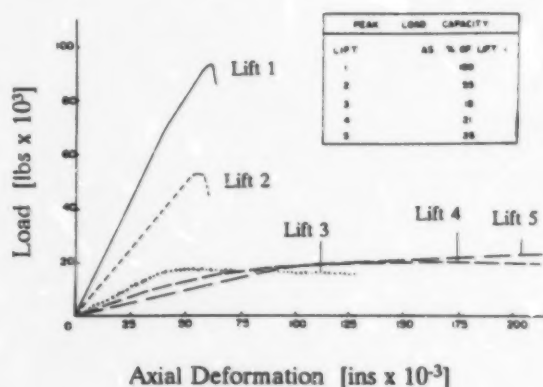


Fig. 4 Failure Curves of 20:1 Cemented Fill Constrained Pillar Models.

Scale Model Failure Mechanism

As small scale rock specimens were used, the mechanism observed was one of the failure of initially intact rock. During loading at the initial 2 lift simulation stage, failure occurred by slabbing along the core surface. During the formation of the third and fourth pillar lifts, extensive fracturing was observed to occur within the zone of the initial two lifts. Conjugate shearing also occurred, reducing the bearing capacity of the pillar to a low level. The upper lifts showed evidence of boundary and internal axial cracks only. In this region a more uniform state of axial compression existed. The upper lift zones were maintained in a more brittle, elastic state.

By the time that lifts 4 and 5 were developed, a reversal in the trend of decreasing support capability with increasing number of lifts was observed. As illustrated in Fig. 4, the overall loading capacity is seen to increase slightly. This strain hardening varied directly with the cement content of the supporting fill, and suggests that the strength and rigidity of fill directly affects the residual pillar strength. The strain hardening behaviour is postulated to occur when sufficient compaction of the basal fractured pillar material, through shear sliding and fill constraint, establishes a stable loading base for the less severely fractured upper pillar zones.

The result of this mechanism is that, as pillar height increased, the upper zones were maintained in a more brittle, elastic state while progressive failure continued to occur at the lower portion of the pillar. With large radial strains into the fill at the lower zones, the presence of the fill resulted in a relatively low but significant post yield pillar strength.

FINITE DIFFERENCE MODELLING

A second modelling method, using a numerical modelling sequence was used to simulate post pillar mining. The objective was to test the ability of finite difference modelling to match the mechanical behaviour of post pillars. A two dimensional finite difference program FLAC, developed by Itasca Consulting Group, Inc. of Minneapolis was used (ITASCA, 1991). This program allows large strains and non-linear material behaviour. The main input parameters for the modelling process include characteristics of the fractured rock mass, the backfill, and their interaction through a cycle of several sequences of pillar excavation and backfill placement.

As discussed in Cundall et al (1988), it is not possible to have sufficient data to model rock masses in a detailed manner. This does not permit the detailed modelling possible in other areas of engineering mechanics. Instead, modelling is used to simplify the real conditions in order to gain conceptual understanding of the processes involved as opposed to making specific predictions.

Modelling Procedure

The rockmass in the area of study is moderately fractured, and is assumed to behave as a continuum with a lower modulus relative to that of the intact rock. The Mohr-Coulomb strength parameters corresponding to the fractured rock mass were based on structural mapping as outlined by Hoek (1988).

A single post pillar was modelled using axisymmetric symmetry. The axisymmetric model consisted of a sequence of 4 m high lifts around an axial pillar having a 2.5 m radius to an outer radius of 13.5 m for an 11 m excavation span. This creates the geometry of a single 5 m diameter post pillar surrounded by a circular room. The left side boundary acts as an axis of symmetry.

Initial stress and gravitational forces as measured *in situ* were applied, and the region to be excavated was removed in successive cuts. Model fill was then added to each previous cut to simulate the cycles of mining and filling.

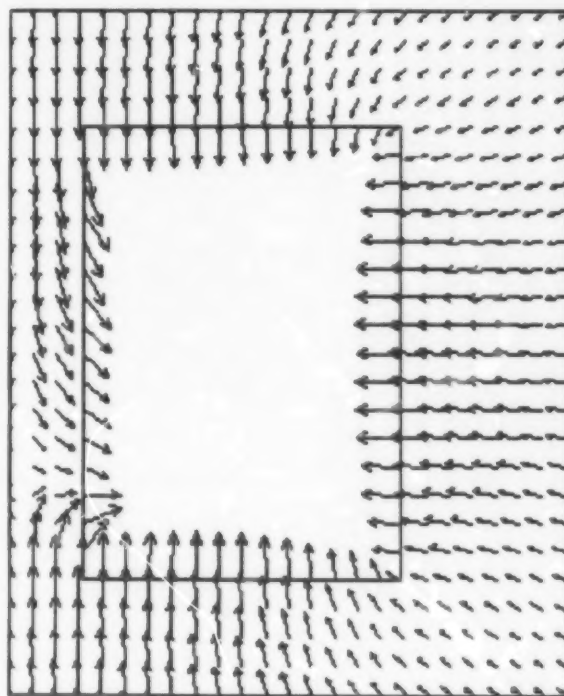


Fig. 5 Displacement vectors after 4 lifts, without fill.

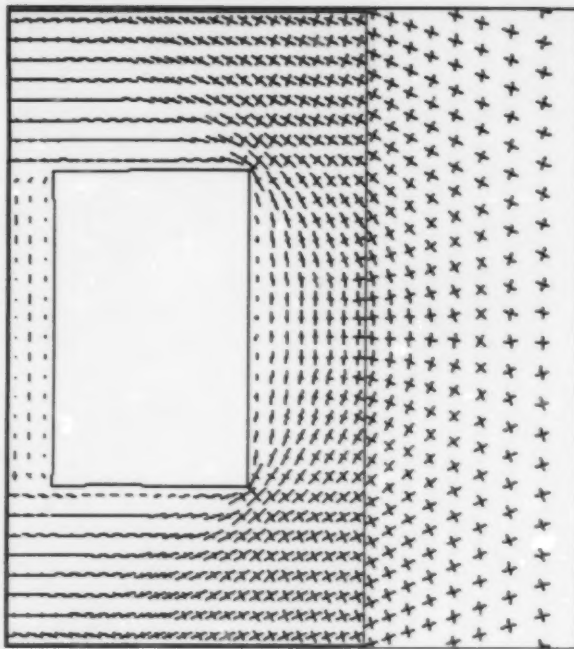


Fig. 6 Distribution of principal stress vectors after 4 lifts, without fill (proportioned to compressive stress, maximum stress vector = 82 MPa).

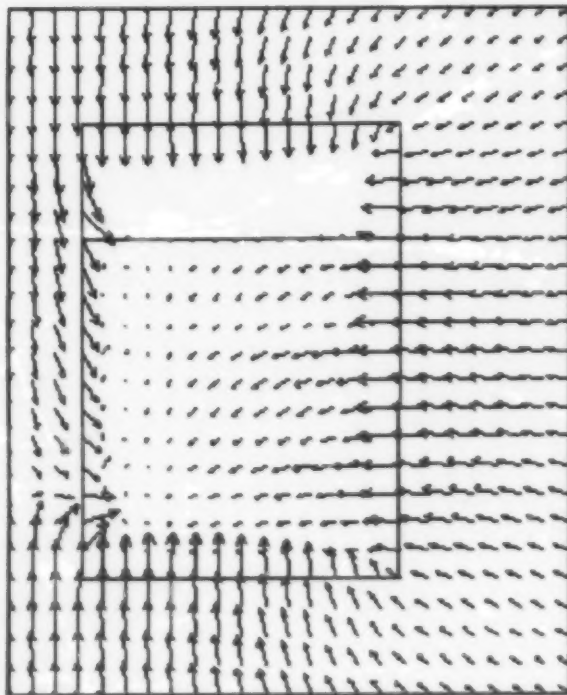


Fig. 7. Displacement vectors after 4 lifts, with fill.

Axisymmetric Model Without Backfill

The axisymmetric model was run over four lifts to a pillar height of 16 meters both with and without the addition of backfill.

High stress levels occurred within the pillar during the first two lifts causing yielding of the pillar. The plot of displacement vectors in Fig. 5 shows a resulting concentration of displacement near the base of the pillar, with the top of the pillar being displaced towards the pillar base.

The principal stress vectors at the nodal gridpoints are shown in Fig. 6. The model indicates the presence of low stress conditions within the pillar, with an arch of low vertical stress above the pillar. The low stress conditions existing within the pillar has resulted from reduction of support applied by the pillar, with subsequent destressing above the pillar and adjacent span. At the completion of modelling, the pillar was shown to be failed and to provide very little support.

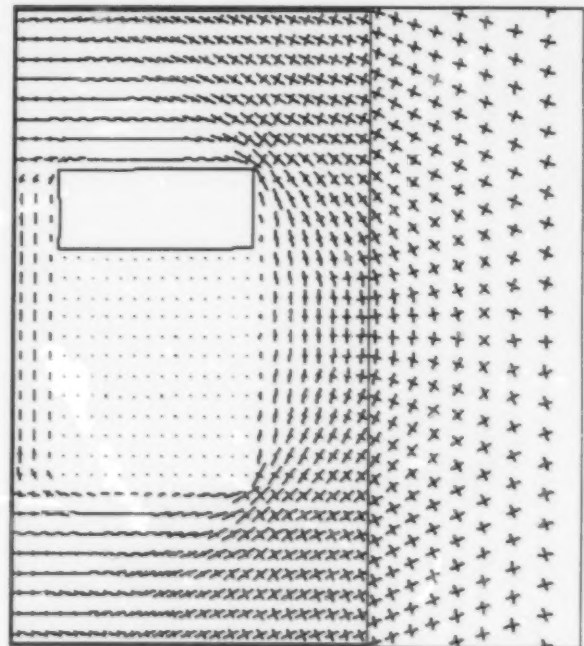


Fig. 8 Distribution of principal stress vectors after 4 lifts, with fill (proportioned to compressive stress, maximum stress vector = 82 MPa).

Axisymmetric model with Backfill

The plot of displacement vectors for this case is shown in Fig. 7. The presence of fill about the pillar was shown to leave the distribution of displacements unaffected compared to the same pillar condition, without fill.

The distribution of principal stresses for the fill confined case is shown in Fig. 8. The principal stresses which develop are shown to increase and to be more uniformly distributed throughout the pillar. An arch of lower vertical stress occurs above the pillar and adjacent span, but to a lesser extent than for the case in which the pillar exists without fill. The increased stress within the pillar indicates maintenance of an increased level of support, resulting in a more favourable stress distribution above the pillar and span than for the unfilled case. At the completion of modelling the pillar was shown to have undergone partial yield but not to have completely failed.

The history plot for maximum principal stress within the pillar is shown in Fig. 9. For both cases of the fill confined and unconfined post pillars, the stress histories are similar until the fourth lift, where without fill the stress declines substantially. A combination of buckling and tensile yielding occurred such that the pillar effectively failed during this lift. This corresponds with the expectation that extensive failure of an actual pillar would occur if fill was not present.

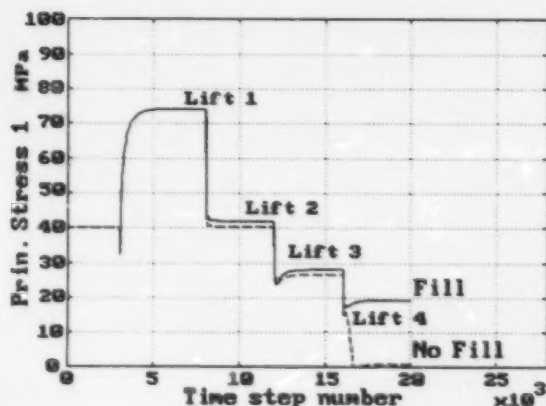


Fig. 9 History of maximum principal stress at 4 m height on pillar centerline over 4 lifts, with and without fill.

This model was extended to a 50 m height to follow the overall trends of stress and deformation response occurring with the fill confined case. The history plot for the maximum principal stress within the pillar is shown in Fig. 10.

The highest stress levels are developed in the pillar during the first lift, with a maximum principal stress developed in the range of 70 MPa. Yield occurs throughout the pillar during the first and second lifts. The maximum principal stress decreases incrementally after the pillar is initially excavated over the first five lifts. Through lifts six to eight, a reversal in the trend of decreasing stress with increasing number of lifts was observed. This occurred due to the transfer of load to the pillar by shear from the settlement of fill along the pillar wall, and to a lesser extent

to increasing lateral support from the fill on the pillar wall. This began to occur after the height of fill exceeded the stope width. With this model, for every 10 m of fill beyond the first 10 m pillar height, the vertical stress at the base of the pillar exceeded that at the top by 1 MPa.

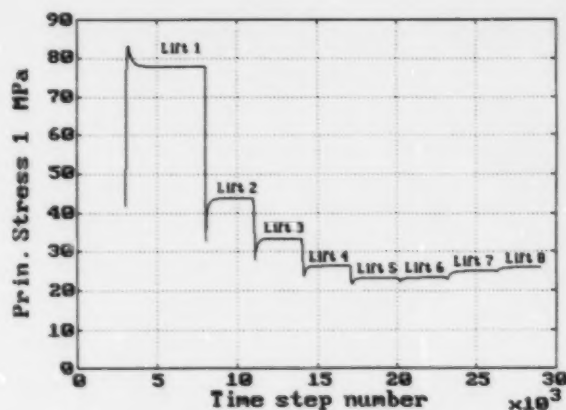


Fig. 10 History of maximum principal stress at the base of a fill confined pillar during excavation to a height of 50 meters.

Numerical Model Failure Mechanism

The actual mechanism of post pillar yielding involves shear along pre-existing discontinuities as well as stress cracking and failure through intact rock. The resulting strength of the pillar is much less than that of intact rock. As the pillar height is increased, some exposed discontinuities will transect the pillar and structurally controlled movement along discontinuity surfaces will occur (Barret et al. 1980). The numerical modelling attempted to account for this by the use of strength parameters based upon the rock mass characteristics in order to model the rock mass as a Mohr-Coulomb elastic-plastic solid.

High stresses occurred within the pillar during the first two lifts resulting in plastic yielding and a concentration of displacements near the base of the pillar. Following exposure of the third lift, the change in pillar geometry and the effect of previous yielding within the lower two lifts effectively reduced the stress within the pillar to below the yield strength of the rockmass.

Without fill, macroscopic pillar failure occurs on the fourth lift due to a combination of tensile yielding and buckling. With the provision of fill confinement and lateral support, failure does not occur.

INSTRUMENTATION RESULTS

It has been found to be very difficult to obtain instrumentation readings from post pillars in deep mines. Problems often occur when fracturing and slabbing within pillars cause a loss of the installed instrumentation. As a result, there are very few instrumentation results available for comparison.

At the Strathcona mine extensometer measurements were obtained from a post pillar developed at a relatively shallow depth of 700 m (Singh and Hedley, 1980). At a 26 m post pillar height the measured contraction between the top and bottom of the pillar was found to be 1.8 cm. When the finite difference model was run using equivalent stress conditions to those estimated at this depth a corresponding contraction of 5.5 cm was obtained. The differing geometries do not allow direct comparison, however, model displacements in the same general range as those for measured instrumentation displacements were obtained.

COMPARISON OF SCALE MODELLING AND NUMERICAL MODELLING RESULTS

The peak strength of the numerically modeled pillar rock was 75 Mpa, compared to 190 MPa used for scale model tests, due to the reduced strength of the large rock mass volumes relative to small intact rock specimens.

The axial strain values through failure for the numerically modelled pillar material was approximated at 0.5 %, compared to about 3 % shown for the physical scale models. The weaker rock mass assumed for the numerical model had a lower strain limit to failure. Also, the physical scale models tested were strained to post yield on each lift, whereas the numerical model pillar yielded mainly on the first two lifts. The resulting pillar stress estimated from numerical modelling pillars was in the range of 20 MPa, compared to 40 MPa for the scale models. With the numerical model the lower zone of the pillar was not as deformed as with the scale model, and instead, smaller strains occurred with a concurrent reduction in stress due to yielding.

CONCLUSIONS

Both physical scale modelling and numerical modelling of post pillar support structures enabled an interpretation of the mechanisms of post pillar yielding and the effect of fill support. Backfill acts to minimize pillar deterioration and prevents complete failure of the pillar from occurring. Within the lower portion of the pillar, fill acts to restrain deformation of the fractured pillar, thus resulting in a low but significant post yield pillar strength. As the pillar yields to a post yield stress level, the effectiveness of the post pillars will not increase at greater depths. However, the main benefit from post pillars is not in providing a support force but in reducing the dimensions of the exposed spans.

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TECHNICAL SESSION 6
ENERGY/MASS TRANSFER

Chair: M. Iwaniw



INDUSTRIAL ENVIRONMENTAL AUDITS

Shawna L. Argue, P.Eng.
Industrial Systems Engineer
Saskmont Engineering Ltd.

ABSTRACT

The following paper discusses the importance of environmental audits to industry. It focuses on why industry should undertake environmental audits, compliance with environmental legislation and conformance with corporate policy and generally accepted good environmental practices. The advantages of environmental audits, how to select an audit team and how to handle the audit findings are also presented.

DEFINITIONS

Environmental Audit

Environmental auditing is "a systematic, documented, periodic and objective evaluation of how well environmental management systems and equipment are performing with the aim of:

- o facilitating management control of environmental practices; and
- o assessing compliance with company policies, including meeting regulated and unregulated requirements."

Audits can also be defined as the verified existence of "adequate...systems...competently...applied." It is important to note that environmental audits are spot checks.

Environmental Management Systems

An environmental management system can be defined as a organizational structure which ensures environmental policy is achieved and maintained throughout an organization. The environmental management system must be an integral part of operations. To be effective it must be important to all levels within the organization from the plant floor to senior management. An effective environmental

management system will be formalized to ensure activities conform with Corporate Policy and comply with legislative requirements. This is accomplished through:

- o an appropriate organizational structure;
- o ensuring accountabilities are understood;
- o project audits or evaluation;
- o communicating environmental concerns within and outside the company;
- o conducting environmental assessments on all capital projects;
- o attention to long term environmental planning; and
- o employee education.

Corporate Environmental Policy

A Corporate Environmental Policy is the company's mission statement towards acting environmentally responsible. Today everyone wants to be seen as being "green" but this is not easily achieved. It takes a conscious effort so that practices become automatic. Policies usually begin with a general statement of the company's philosophies. These generalizations are followed by the specific policy goals.

It is imperative that the Corporate Environmental Policy be communicated to all

employees. It should also be written in a clear, concise format which can be easily posted. Also, managers should be supplied with guidelines for the implementation of the policy as it pertains to their operation.

Corporate Environmental Procedures

Corporate Environmental Procedures are operating guidelines which have been set by the company. They may include:

- o Spill Response Procedures;
- o Emergency Procedures (ie: fire, first aid); and
- o Public Complaint Policy.

WHY UNDERTAKE AN ENVIRONMENTAL AUDIT???

As the complexity of environmental legislation increases and the penalties for non-compliance become more severe, environmental audits can be utilized to evaluate an industry's present operations.

Environmental audits are also utilized by companies to show due diligence. It is no longer acceptable by regulatory agencies or the public to continue operating with a "fix it when it happens" attitude. By initiating an audit program, industries are being proactive, identifying their activities, regulated or unregulated, which may pose a risk to the environment. The organization can then utilize this information for planning and management to ensure they meet corporate environmental objectives. Audits can assist in prioritization of environmental problems, thus allowing resources to be directed to the most important issues.

The two main reasons for industry to initiate an environmental audit are:

- o to determine whether or not present operations are conducted in an environmentally acceptable manner and in compliance with Federal, Provincial or local legislation and Corporate Environmental Policy, where applicable; and
- o to determine possible long-term risks. This includes assisting management in assessing operations in the area of:
 - . regulatory compliance;
 - . environmental contamination;
 - . health and safety;
 - . Corporate Policy conformance;
 - . financial planning;
 - . risk analysis; and
 - . reviews of existing legislation or anticipated legislative trends.

Environmental audits can improve and maintain a company's public image. The information provided by the audit permits a company to be proactive, anticipating problem areas and remedying these areas. An organization can constantly improve their

practices, thus leading the way rather than being driven by regulatory requirements.

Environmental audits are of particular importance to manufacturers as they can reveal cost saving opportunities. These may include changing or decreasing raw materials which may result in reduced waste production or may be an alteration in production flow resulting in more efficient operation.

Audits demonstrate due diligence, thus minimizing risks for the corporation and its officials. This is important with the growing trend towards increasing fines and imprisonment of company officials for violating environmental regulations. Audits are useful in identifying areas of non-compliance and risk. Early discovery of these activities, and attention given to correcting these areas, will reduce the likelihood of penalty.

TYPES OF AUDITS

There are several different types of environmental audits. Each type of audit requires different expertise and varying levels of detailed investigation to achieve the objectives.

Types of environmental audits include:

- o Environmental Compliance; Regulatory
- o Property Transfer;
- o Waste Minimization/Management (RRR);
- o Environmental Management Systems;
- o Corporate Environmental Policy and Procedures Conformance; and
- o Risk and Financial Liability Assessment.

An audit can examine one, two or all of these topics. Audits can be effectively conducted by one person, a small team or a team of five or six members depending on the scope and complexity. Audits may take the form of "head office" paper/file reviews to in-depth site evaluations. Audits may take two or three days to complete or ten to twelve weeks including one to two weeks on-site.

Duration is determined by the type of audit to be undertaken, the size of the facility being reviewed and the specific objectives and scope, set prior to audit implementation. It is important to remember that an audit is a snap shot of the facility's operations...it is not the whole movie. Comprehensive audits will provide a "picture" of what the audit team sees at the time of the audit. It is inappropriate for the team to speculate what the situation was the day before the audit or what it will be the day after the audit.

ENVIRONMENTAL AUDIT TOPICS

An audit can evaluate performance in all or some of the following areas:

- o Environmental Management;
- o Water and Waste Water Management;
- o Air Quality Management;
- o Solid Waste Management;
- o Hazardous Materials and Waste Management;
- o Occupational Health and Safety;
- o Spill and Emergency Preparedness; and
- o General Environmental Practices.

Which of these categories will be reviewed depends on the type of audit, the facility being reviewed and the specific audit objectives and scope.

OBJECTIVES OF AN ENVIRONMENTAL AUDIT

Environmental audits will provide a periodic, systematic and objective review of a facility's operations and practices. Specific objectives of an audit program are the specific goals or targets the program aims to achieve. Objectives will vary from site to site, as will the scope of the audit.

Audits can be developed to achieve any or all of the following objectives:

- . verify environmental compliance;
- . evaluate the effectiveness of environmental management systems; and/or
- . assess risks from regulated or unregulated practices.

It is pertinent that organizations show environmental awareness in all of their practices. Audits can measure and evaluate a facility's level of awareness while highlighting strong points and weaknesses.

HOW TO SELECT AN AUDITOR

There are two philosophies of selecting audit staff:

- o appoint internal staff to conduct audits; or
- o hire an external team to conduct audits.

It is estimated that over 90% of Internal Environmental Auditors are part of the company's "Environmental Affairs" department. This has the advantage of having people in the department with access to and familiarity with the subject matter being audited. The disadvantage of this approach is the potential for conflict of interest. Team members may be auditing programs in which they had other involvement. Also, the requirement of independence, ensuring that auditors are "one step removed" from the activities being audited, may not be met.

Another department which may house the audit team is Operations or Production. This will reinforce the concept that Operations personnel are responsible for environmental management. Again, the problems of conflict of interest and independence may arise.

The legal department may occasionally be selected to house the audit team. The legal department may be selected:

- o where problems of potential significance are suspected;
- o where the company is under particular scrutiny from regulatory agencies; and
- o where there is strong concern on the part of management to maintain confidentiality.

However, housing your audit program in the legal department may result in the audit becoming a legal tool rather than a management tool. Therefore, audits are generally managed under the legal department only if:

- o it is the type of review that would routinely be handled by the legal department; or
- o it is a special situation that warrants particular legal scrutiny.

The Internal Audit department is sometimes chosen to house the environmental audit program. The advantages of this choice are that there is a high level of independence and it utilizes the extensive experience that internal auditors have in the discipline of auditing. However, it is important to ensure that the individuals conducting the audit also possess the technical knowledge and expertise necessary.

These in-house options are only achievable in large organizations. Many smaller operations do not have the in-house expertise to conduct an environmental audit. For this reason, they turn to external auditors for assistance. As well, many large corporations are choosing external auditors to ensure that there are no conflicts of interest and to meet the requirement of independence.

Whether choosing in-house staff to form an audit team or selecting a third party consultant to conduct your audit program, it is necessary to select a team with the credibility and capability to get the job done. It is essential that environmental audits be conducted by personnel who collectively have the skills and experience necessary for the type of audit that is to be conducted. The types of expertise required include:

- o technical training and experience
- important in auditing the facility's compliance with regulatory requirements but especially necessary when assessing the extent to which

the site, in the judgment of the audit team, conforms to "good management practices" or "good environmental practices".

- o understanding of auditing
 - team members must have an understanding of basic audit theory and procedures and the experience to apply it. Some audit consultants have developed audit procedures manuals to which all audit team members must adhere.
- o knowledge of the regulations
 - each auditor should have a working knowledge of the regulations, criteria and standards appropriate to the specific topics he/she is auditing.
- o familiarity with the operations
 - someone on the team should have a general familiarity with the type of operations being reviewed and the issues likely to be encountered, usually the Audit Team Leader.

It has been said that when purchasing environmental services, one should really be looking at it as though they were going in for heart surgery, where they would be looking at the qualifications of the surgeon. Ask yourself:

"Who are the active players and what are their qualifications?"

"How are they going to perform their work and what's their track record?"

An audit team can consist of:

- o Engineers;
- o Hazardous Waste Specialists;
- o Agrologists;
- o Biologists;
- o Air Quality Specialists;
- o Hydrogeologists;
- o Chemists;
- o Technologists; and
- o Occupational Health Specialists.

This team can be mixed and matched according to the type and size of the audit being undertaken.

BENEFITS OF AN ENVIRONMENTAL AUDIT

Environmental audits are viewed as a management tool. They will bring into focus environmental issues that require attention. Management can then develop an action plan and place projects into the planning and budget process.

Action plans are developed from the audit findings and observations. Findings

are prioritized, individual responsibility assigned and a target deadline set. The specifics of how to address the finding is not, necessarily, outlined. This activity can be left up to the individual responsible for the task.

Audits will allow facilities to bring practices into compliance with existing legislation as well as prepare for upcoming changes.

Environmental audits can point out risks that companies never knew existed. For example, the transferring of dangerous goods (hazardous wastes) to a waste disposal company does not absolve the generator of the responsibility for proper disposal. Facilities should make every effort to know the final destination of the waste and how it will be handled. They should also ensure that it reaches the designated destination.

The benefits of environmental auditing are many and include:

- o identification of compliance issues and risks of non-compliance;
- o improved environmental management systems;
- o sensitization of senior management and operating/production/plant staff;
- o better communication and enhanced general understanding of facility operations;
- o improved management structure;
- o cost savings and efficiency improvements;
- o improved awareness of environmental regulations and hazards; and
- o improved manufacturing efficiency.

FREQUENCY OF ENVIRONMENTAL AUDITS

Once the initial audit has been completed and an action plan set out, the facility should undergo "mini" audits every six months to one year to gauge performance against the action plan deadlines and to assess any new issues that may have arisen. Follow up audits would not require a great deal of effort. They would be similar to a routine Occupational Health and Safety inspection. A full scale audit should be conducted once every three to four years.

LEGISLATION AND PERMITS

In evaluating regulatory compliance, one must assess practices against what is required in legislation and permits.

Legislation consists of Federal and Provincial Acts and Regulations and Municipal Bylaws. They are general rules of operating - "do's and don'ts".

Permits or Ministerial Orders are usually specific to the facility and may have additional requirements for the facility or

may exclude the facility from some of the regulatory requirements.

Which legislation is reviewed during an audit depends on what is applicable to the operation being audited. Some operations may be affected by several acts or regulations while others may only need comply with one or two.

Following is a list of the legislation most frequently of concern. It is not an exhaustive list.

Federal

- o Canadian Environmental Protection Act (CEPA)
- o Fisheries Act
- o Transportation of Dangerous Goods Act
- o Hazardous Products Act (WHMIS)
- o The Environmental Contaminants Act

Provincial (Saskatchewan)

- o Clean Air Act
- o Dangerous Goods Transportation Act
- o Environmental Management and Protection Act (EMPA)
- o Environmental Spill Control Regulations
- o Hazardous Substances Regulations
- o Municipal Refuse Management Regulations
- o The Fisheries Act
- o The Ground Water Conservation Act

Municipal

Bylaws will vary from municipality to municipality. Ones to look for are those pertaining to:

- o noise;
- o zoning;
- o use of sewer systems;
- o fire code; and
- o emissions control.

The typical Permit to Operate and Ministerial Orders reviewed include:

- o Permit to Operate under The Clean Air Act
- o Permit to Operate a Waste Disposal Grounds under the Environmental Management and Protection Act
- o Radioisotope License
- o Permit for Sewage or Industrial Effluent Works
- o Approval to Operate Works under The Water Corporation Act
- o Waste Dangerous Goods Generator/Receiver/Carrier Number
- o Permit to Store Hazardous Substances under The Hazardous Substances Regulations

Not all of these permits will be applicable to every facility. Some facilities may not require any permits, although this would be rare. Other facilities may require permits different from those listed.

CONCLUSIONS

In conclusion, an environmental audit should be conducted if an organization meets one, some or all of the following:

- o transferring some property;
- o unsure of compliance status;
- o suspect that your operations may be causing harm to the environment;
- o user, generator or receiver of dangerous goods;
- o generally good environmental practice.

Environmental audits can provide facilities with:

- o a verification of environmental regulatory compliance;
- o an evaluation of the company's environmental management system;
- o an assessment of risks of past and current practices;
- o an identification of cost saving opportunities; and
- o an improved public image.

Environmental audits should be conducted regularly to demonstrate due diligence. That is, that the organization exercised all reasonable care by establishing a proper system to prevent non-compliance and is taking reasonable steps to ensure the effective operation of the system.

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MODELLING GROUNDWATER FLOW AND SOLUTE TRANSPORT USING THE MOTIF FINITE-ELEMENT CODE

Yee-Chung Jin
University of Regina, Regina, Saskatchewan

Tin Chan
Whiteshell Laboratories, AECL Research, Pinawa, Manitoba

Dankyi N. Osei
University of Regina, Regina, Saskatchewan

and

Stephen Butt
Whiteshell Laboratories, AECL Research, Pinawa, Manitoba

ABSTRACT

The Whiteshell Laboratories of AECL Research, has developed a finite element code called MOTIF for modeling the inter-related processes of groundwater flow, heat and contaminant transport in plutonic rocks. The code has been employed in predicting contaminant transport, first in a hypothetical aquifer and second in a field scale aquifer. In the first case involving the modelling of a conservative solute transport in a hypothetical aquifer using identical results were obtained when compared with the results of an independent mathematical modeling of the same problem. The second was an investigation of the migration of a radioactive tracer in a model of the Twin Lake aquifer. The tracer plume contours showed similar shape and pattern as the field data at the initial stages, but the migration pattern showed some deviations after a certain time. In general the MOTIF code has been demonstrated to provide reasonably accurate solutions to field scale problems of groundwater flow and solute transport in heterogeneous media.

1. INTRODUCTION

Simulation of groundwater flow and associated heat and contaminant transport in the subsurface environment, using analytical model is possible only under idealized conditions. This has led to the development or adoption of various numerical methods in this field of engineering.

As part of the research and development with Canadian Nuclear Fuel Waste Management Program, AECL (Atomic Energy of Canada Limited) has developed a special computer code called MOTIF. The MOTIF (Model Of Transport In Fractured/Porous Media) is a three-dimensional finite element code for solving the coupled equations of groundwater flow, conductive and convective heat transport, and radionuclide transport (Guvanasen 1984 and Chan et al. 1987, Chan, 1989).

The numerical accuracy of the MOTIF code has been verified by comparison with analytical and independent numerical solutions

(Chan et al. 1989). The paper consists of two parts. The first part presents a comparison of the code's numerical results with a mathematical model results of a nonreactive solute transport in a hypothetical unconfined aquifer. The second part, outlines the use of the MOTIF in simulating a field trace test. This involves the simulation of the migration of a radiotracer in a two-dimensional model of the Twin Lake aquifer. This objective was accomplished by using an extensive data base from an in situ natural gradient dispersion test conducted at the Twin Lake aquifer in 1983 by AECL's Chalk River Nuclear Laboratories (Molyneux and Willis, 1987). The MOTIF was used to simulate this trace test and the predictions are compared with the field data.

2. GOVERNING EQUATION

The conventional convection-dispersion equation for solute transport in a saturated

porous media (bear, 1972) can be written as

$$\frac{\partial}{\partial t}(\theta C) + \frac{\partial}{\partial x_i}(\theta u_i C) - \frac{\partial}{\partial x_i}(\theta D_{ij} \frac{\partial C}{\partial x_j}) = 0 \quad (1)$$

in which θ is porosity, u_i is the i -th component of interstitial velocity, and C is the contaminant concentration. In equation (1), D_{ij} is the hydrodynamic dispersion coefficient tensor which, for an isotropic medium, can be expressed as

$$D_{ij} = \alpha_T u \delta_{ij} + (\alpha_L - \alpha_T) \frac{u_i u_j}{u} + \tau D_0 \delta_{ij} \quad (2)$$

where u is the magnitude of the linear water velocity, α_L is the longitudinal dispersivity, α_T is the transverse dispersivity, D_0 is the molecular diffusion coefficient in free water, τ is the scalar tortuosity factor, and δ_{ij} is the Kronecker delta.

In MOTIF the Galerkin method of weighted residuals is used in conjunction with the finite-element method to solve the solute transport equation.

3. A SIMULATION OF A NONREACTIVE SOLUTE IN AN UNCONFINED AQUIFER.

A hypothetical aquifer (Sudicky, 1989) was used for modeling the movement of a conservative solute in a unconfined aquifer. The hydraulic parameters in this hypothetical aquifer were chosen to be suitable for testing the performance of a numerical model and therefore this test case was used to assess the capability of the MOTIF code for modelling groundwater pollution problems. The aquifer has longitudinal and vertical dimensions of approximately 250m and 6m respectively. The aquifer material comprises of fine grained silty-sand within which is located a discontinuous 2 metre thick medium grained sand layer. The length of the medium grained sand layer at the left and right ends of the aquifer measures 120m and 70m respectively. Figure 1 shows the aquifer stratigraphy.

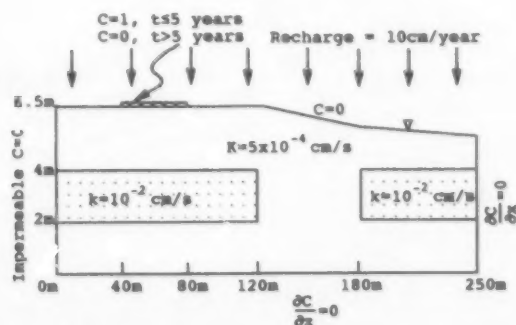


Figure 1. The model stratigraphy and groundwater flow parameters. (Sudicky, 1989)

The two geologic units are isotropic and their hydraulic conductivities, porosities, dispersivities and diffusion coefficients are shown in table 1.

Table 1 Transport Parameters used in model 1

	k	θ	α_L	α_T	D_0
silty sand	5×10^{-6} m/s	0.35	0.5m	0.005m	1.34×10^{-5} cm^2/s
medium grained sand	1×10^{-4} m/s	0.35	0.5m	0.005m	1.34×10^{-5} cm^2/s

The top of the aquifer was specified as a free water table with an annual recharge of 0.1m. A constant hydraulic head of 5.375m was specified at the right edge of the aquifer. The left and bottom of the aquifer were specified as impermeable. The boundary conditions for the solute transport are as indicated in figure 1. The bottom and the right boundaries were specified to have a concentration gradient of zero normal to them that is a second type boundary condition. The concentration is 1.0 units at the source of injection for a five year duration and it is assumed to be zero outside this source. The contaminant (a nonreactive solute) was injected into the aquifer along the water table between 40m and 80m from the left edge.

Initially, a groundwater flow model was created with the top boundary of a constant elevation of 5.375m and recharge of 0.1m/year. The model consisted of 2600 quadrilateral elements as shown in figure 2. The flow was simulated by MOTIF assuming steady state condition. Nodes on the top of the mesh was adjusted to match the calculated groundwater table. The nodes beneath each surface node were subsequently adjusted proportionally to maintain equal nodal spacing. This process was iterated until the difference of the top nodal elevations and the computed water table were 0.01% of the elevations.

The model geometry for the flow simulation was incorporated into the the solute transport model. Ninety-two time steps

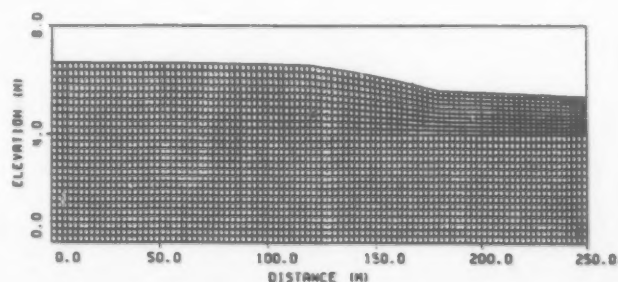


Figure 2. The mesh used in the hypothetical model.

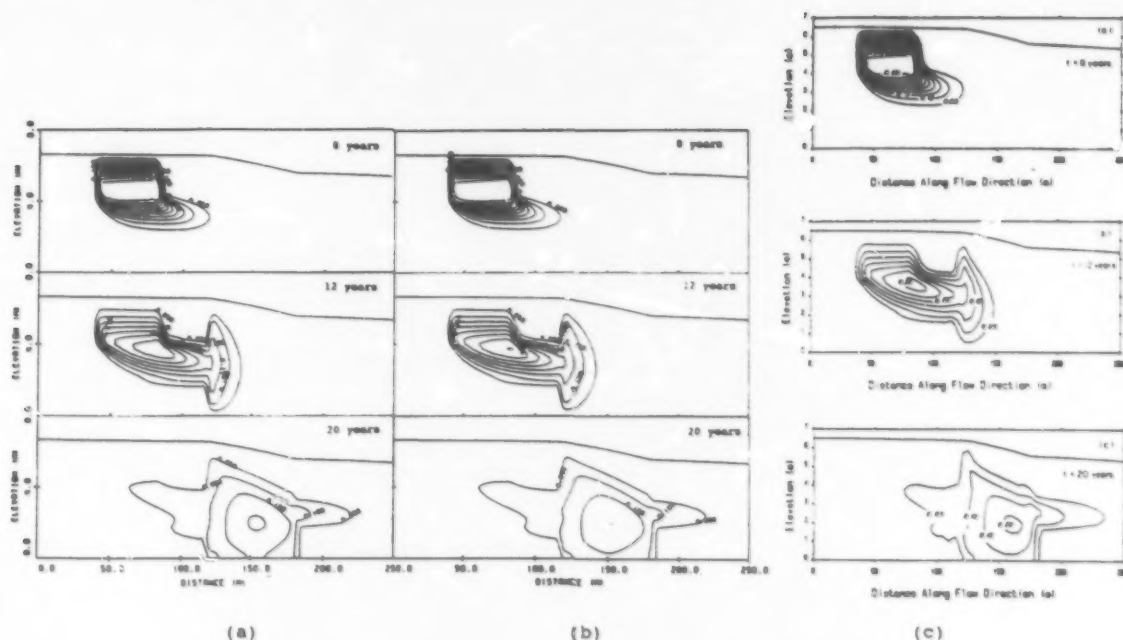


Figure 3. The contaminant plume contours. (a) MOTIF result (b) MOTIF result (with artificial diffusion). (c) Sudicky's result

were selected for the contaminant transport model beginning with 100 seconds and increasing to an interval of 0.25 years. The element Peclet number (a dimensionless constant which relates the groundwater flow velocity, the maximum element width and the contaminant dispersion) was approximately 5.

The model was run and the contaminant plume contours generated by the model for eight, twelve and twenty years are presented in figure 3(a).

3.1 Discussion

The plume initially moved downward slowly until it reached the left sandy layer where it flowed laterally. This lateral flow is due to the groundwater velocity in the sandy layer being higher than that of the surrounding region. The leading edge of the plume spreads out as it re-enters the silty sand region, but converges again as it enters the right sandy layer. The final result compared very well with Sudicky's Laplace Transform Galekin finite-element modelling results of the same problem (see figures 3(a) and 3(c)).

Though the results were in good agreement with existing results, there appeared to be numerical oscillations since negative concentration values appeared in the initial model output. The aquifer model was re-solved by introducing artificial diffusion to see if the results would improve. This latter result was encouraging as the negative concentration values were lower and the plume contours appeared smoother than before. In the twelve

year contours, the amount of dispersion apparently decreased slightly as indicated by the presence of a small region of 0.4 concentration at the core of the plume. This was absent in the original model as seen in figure 3(b). For the 20 year contours, the dispersion has decreased apparently as the 0.20 contour is smaller and the 0.15 contour does not reach the model boundary as before. For all time steps, the maximum negative concentration value was less in the latter case than the initial model where there was no artificial diffusion.

4. THE TWIN LAKE AQUIFER

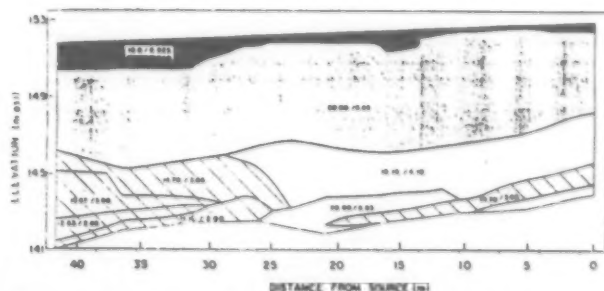
An in situ natural gradient dispersion test was performed at the Chalk River Nuclear Laboratories in 1983 using Radioiodine (^{131}I) and tritium (^3H) (Killey and Moltyaner 1987). The experimental installations at the tracer test site include multilevel sampling devices and dry/access observation wells (Moltyaner and Wills 1987). The measurements were obtained using a technique known as through-the-wall measuring technique. Some of the ^3H and ^{131}I concentrations were measured using isolated multilevel solution sampling devices. The vertical dimension of the screened interval of multilevel samplers used at the Twin Lake site is 1m. The samplers are spaced at 5m in the direction of flow and at 1m in the perpendicular direction. The fully penetrating dry observation wells used for this purpose are also spaced at 5m longitudinally and 1m laterally. During the test, Radioiodine (^{131}I) was injected into the aquifer and its migration monitored in the dry observation wells using NaI detector. The

data from this experiment has been simulated using MOTIF.

The aquifer stratigraphy showing the horizontal and vertical hydraulic conductivities are shown in figure 4. This hydraulic conductivity distribution has been estimated by grain-sized analysis of cores (Moltyaner, private communication). The large hydraulic gradient along the water table and the high hydraulic conductivities of the sand and gravel aquifer resulted in relatively high groundwater velocities of approximately 2×10^{-6} m/s. As a result, a nodal spacing of 0.1m was selected to reduce the element Peclet number. The resulting finite element mesh has approximately 43000 nodes and the Peclet number was 10. In order to perform the simulation in a reasonable time, the model was broken down into three separate smaller sub-models which were run separately under the same conditions. Figure 5 shows these submodels. Each model has a variable mesh size, one half having nodal spacing of 0.1m and the remaining half having nodal spacing ranging from 1.0m to 2.0m. To ensure continuity of flow of the tracer plume, from one sub-model to another, the mesh in each model was designed to have a 50% overlap of finely discretized mesh with the adjoining model in the sequence. This allows the tracer plume to be continuously modelled using an optimal nodal spacing.

The boundary conditions for the groundwater flow were determined from the hydraulic heads measured from the piezometer readings. Nodes along the left and right boundaries of the model were assigned hydraulic heads approximated from the hydraulic head contours drawn from the piezometer readings, whereas the nodes along the water table were assigned heads equal to their elevations. The bottom of the model was assumed to be impermeable and the porosity of the aquifer material fixed at 0.38 since there was not enough information to determine the porosity. These conditions were used to develop the groundwater flow model.

Using the velocities calculated by the flow model a two-dimensional MOTIF transient solute transport model was used to simulate



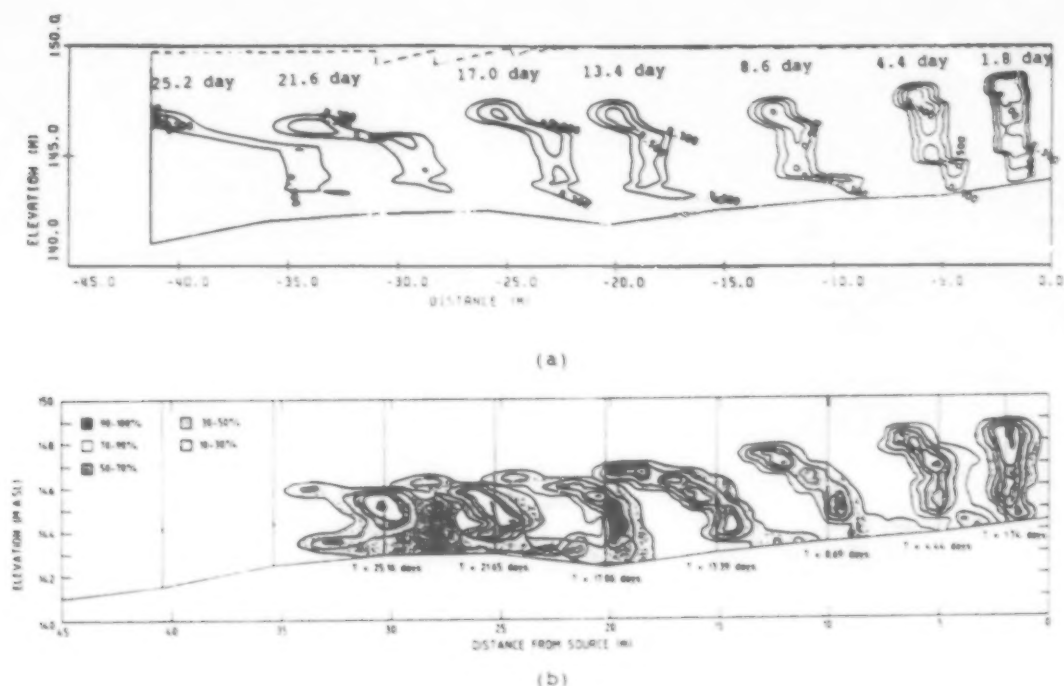


Figure 6. The profile view of the tracer plume migration. (a) Moltanier's field data (b) MOTIF result.

developed acceptable shape at the initial stages, the plume shape began to show slight deviations from the measured shape after about 8 days. It can also be seen that according to the model, the tracer near the bottom section of the plume in contact with the bedrock surface moved slower than the top section. This continued with time and after about 18 days, the top front of the plume was far ahead of that of the bottom section. This was an indication of the velocity at the top increasing at a higher rate with time than at the bottom. Although the measured tracer plume also showed a similar trend, after about 24 days, there was large deviation from the measured shape. This happened after the plume has entered the coarse aquifer material.

The deviation that resulted may be due to a number of factors among which are: (1) changes or modifications made to the aquifer stratigraphy to ensure that constant hydraulic conductivity is maintained in the sub-region; (2) the irregular surface of the bedrock; (3) the selection of some of the flow boundary conditions which could probably cause the higher flow velocity near the top. The flow boundary and initial conditions could be a problem because there was not enough information to determine the best conditions. The deviation could also be due to an inaccurate choice of dispersion coefficient. In general, the result is fair considering the fact that rather simplified aquifer geometry and parameters were used and no calibration was attempted to match the observed tracer plume migration by adjusting the aquifer parameters.

5. CONCLUSIONS

The results of the first model compare well with Sudicky's model indicating the reliability of the MOTIF code. MOTIF has also demonstrated to be capable of handling a field scale problems in a heterogeneous aquifer. It can therefore be used to model flows in heterogeneous aquifer environment. This modeling exercise also suggest that the lower the element Peclet number the more stable the results.

The Twin Lake aquifer model yielded a fair simulation. The deviation in the tracer migration pattern may be attributed to the following factors: i) the boundary conditions for the groundwater flow may not be accurate hence the resulting high groundwater velocity and the corresponding fast plume movement; ii) the nonuniform flow may be due to the irregular bedrock surface; iii) the simplification of the aquifer stratigraphy; iv) the contaminant transport boundary conditions and non-uniform dispersion coefficient.

The results of this model could possibly be improved by maintaining the aquifer stratigraphy without any modification. Expanding the mesh in the model into a three-dimensional case to give the contaminant an extra dimension for transport and dispersion may improve the result. Because of larger number of elements will be needed in a three dimensional model, other numerical technique may have to be introduced in MOTIF to handle larger Peclet number problems.

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FIBER OPTIC VORTEX SHEDDING FLOWMETER

A. H. Morshed and D. Irvine-Halliday

Electrical and Computer Eng. Department,

Univ. of Calgary, Calgary, Alberta T2N 1N4

ABSTRACT

Intensity modulated sensing of fluid flow rate is achieved by using single-mode optical fiber tapers to detect the shedding frequency in a vortex shedding flowmeter. Minimum flow blockage occurs when the fiber taper is used as a shedding strut. However, due to the fiber mechanical flexibility, a separate shedding strut would be needed. Both configurations were investigated under multi-mode, and single-mode operation.

INTRODUCTION

Developing an optical fiber fluid flowmeter for applications in industrial process monitoring and control is a subject which has received considerable attention. Flow meter configurations employing fiber optic sensing of the flow effects have been reported [1-8] and of these the vortex shedding flowmeters have the potential of low cost, easy fabrication and negligible flow blocking. Those were implemented using phase modulation of the optical signal with either a multi-mode [5] or a Fiber Fabry-Perot [7, 8] interference detection scheme. An intensity modulated vortex shedding meter using fiber microbends was also demonstrated [6]. A completely different sensor is reported herein where a fiber taper is used to detect the shedding frequency using a simple mechanical arrangement and intensity detection scheme.

OPTICAL FIBER TAPERS CHARACTERIZATION AND MODELING

Fiber tapers were made from a single-mode communications fiber with a cutoff wavelength of 1.245 μm and mode field and cladding diameters of 9.5 and 125 μm , respectively. These tapers were measured and characterized for their relative transmission versus bending at 0.633 μm and 1.3 μm where the fiber acts as a multi-mode and a single-mode light guide, respectively. Fig.1 gives the transmission characteristics of a fiber taper at both wavelengths. At 0.633 μm the normalized frequency at the taper waist is 1.15. The higher order core-

guided modes of the fiber couple completely to the cladding-guided modes which undergo severe mode mixing within the taper when passing through the taper section with a normalized frequency of 2.4. The fundamental mode participates only partially in the coupling process taking place at the taper waist which is then modulated by bending. At 1.3 μm , the normalized frequency at the taper waist is 0.56 and deeper modulation of transmission with bending is observed. At values of the normalized frequency less than unity the core-guided mode of the fiber has a field profile coincident with that of the lowest order cladding-guided mode and all core guided power in the fiber transfers to the cladding modes [9, 10]. Moreover, since the periodicity of coupling with bending angle is different for each guided mode, the transmission characteristics in multi-mode operation has more minima and maxima in the same bending range than in single-mode operation. Single-mode fiber tapers with waist diameters giving a normalized frequency less than unity thus give more transmission modulation and a wider range of monotonic variation with bending, thereby allowing for higher bending-sensor sensitivities and range of operation. These tapers were modeled for their characteristics using local normal modes and mode mixing within the waist regions [11, 12]. The solid line of Fig.2 gives the results of computations using this model as compared to experimental taper characteristics. The dashed line gives the computed transmission characteristics when the medium external to the fiber is water having a refractive index of 1.333 instead of 1.0 for air. For the use of the taper in a vortex shedding flowmeter it would be immersed in the fluid and respond to lift forces of the flow.

THE VORTEX SHEDDING FLOWMETER

A prototype vortex shedding flowmeter was constructed and tested for water flow rate measurement using a 1" internal diameter pipe flow rig with a 1 cubic meter tank and 0.75 horse-power pump. The flow was controlled by needle valves and indicated by a rotameter over a range of average velocities between 0.3 and 3 m/s based on rotameter indication. Calibration of this rig by discharge measurements indicated an average error of 0.032 m/s in velocity calculation throughout the range. The fiber taper spanning a diameter of the flow pipe was installed into a test section made of plexiglass. The transmitted optical power through the taper was detected, amplified and its modulation spectrum displayed. Fig.3 gives the modulation spectrum at average flow velocity of 0.77 m/s, with 1.3 μm wavelength. Fig.4 displays the detected frequency peaks with the fiber as the strut at 0.633 and 1.3 μm wavelengths, as compared to the shedding frequencies of a rigid 125 μm rod, which is the outer diameter of the fiber used. The solid line in this figure is obtained from Roshko's formulae for the vortex shedding frequency past a rigid circular cylinder over the laminar and transitional ranges of vortex shedding [13]. The peaks barely follow the shedding frequencies of such a strut due to the flexibility and inhomogeneity of the fiber taper. At 0.633 μm , higher order harmonics are also detected thus preventing a unique determination of the shedding frequency. The distribution of lower order peaks around the theoretically expected frequencies, particularly in the laminar vortex shedding range, suggests that the mechanical properties of the taper as a shedding strut has a significant effect on the ultimate oscillation frequency of the compound wake-fiber system [14].

Using the fiber taper as a secondary strut, past a rigid primary shedding strut, enabled an unambiguous detection of the shedding frequencies throughout the investigated range of flow. With a steel rod of 750 μm diameter as the shedding strut and the fiber taper installed 3mm downstream, the frequency peaks displayed in Fig.5 are obtained at 0.633 μm . Shedding frequencies corresponding to both the primary strut and the fiber can be observed as well as harmonics. Fig.6 shows that at 1.3 μm only shedding frequencies of the primary strut are detected. Fig. 7 and 8 give the intensity modulation spectra at an average flow velocity of 1.31 m/s for both wavelengths. In both cases, the peaks corresponding to the shedding frequencies of the primary strut follow the theoretical frequencies [13] to a very large extent. Installing a different taper past the strut does not have a significant effect on shedding frequencies, as shown in Fig.9.

CONCLUSION

A fiber optic vortex shedding flow meter was demonstrated using fiber tapers to detect the shedding frequencies in a simple mechanical arrangement and utilizing direct intensity-modulation detection. A prototype of the flow meter was realized and investigated for water flow rate measurement under multimode and single-mode fiber operation. Due to the fiber mechanical flexibility a separate rigid shedding strut is needed, and using such a primary shedding strut, shedding frequencies following closely those theoretically expected were detected throughout the investigated flow range.

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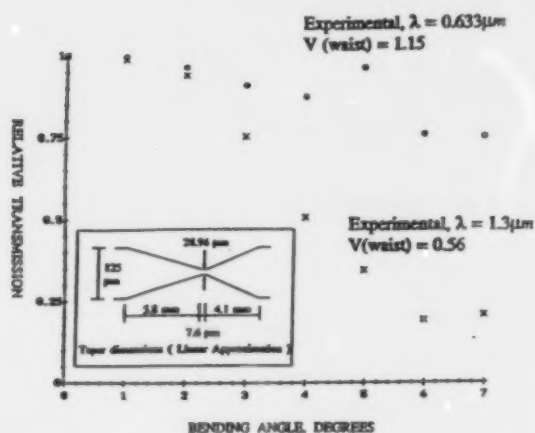


Fig.1. Experimental relative transmission versus bending for the fiber taper shown in the insert at 0.633, and 1.3 μm .

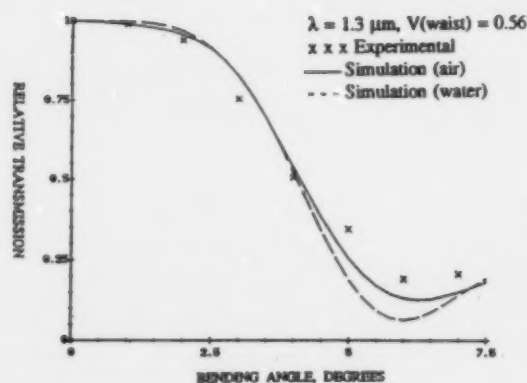


Fig.2. Relative Transmission versus bending of the fiber taper of Fig.1, at 1.3 μm .

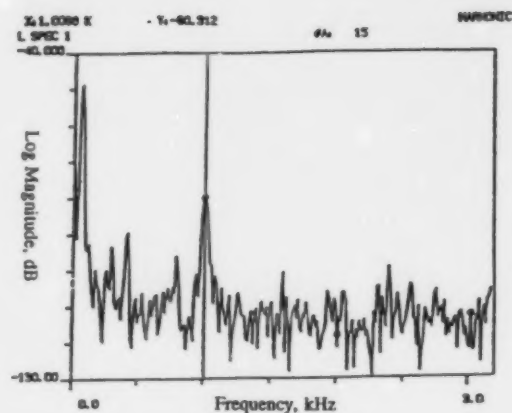


Fig.3. Spectrum of the intensity modulated signal with the fiber taper as a strut, at average flow velocity of 0.77 m/s and 1.3 μm light wavelength.

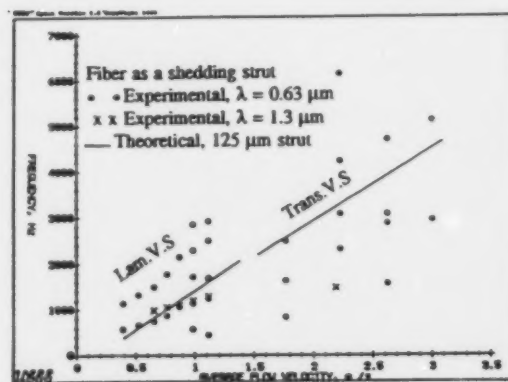


Fig.4. Detected frequency peaks with the fiber taper as a strut, at 0.633, and 1.3 μm light wavelength as compared to the shedding frequencies of a 125 μm strut.

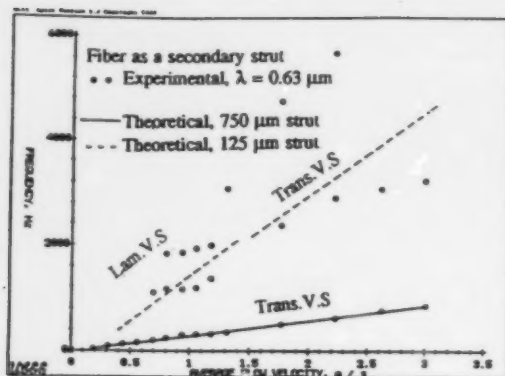


Fig.5. Detected frequency peaks with the fiber as a secondary strut, past a rigid shedding strut of 750 μm diameter, at light wavelength of 0.63 μm .

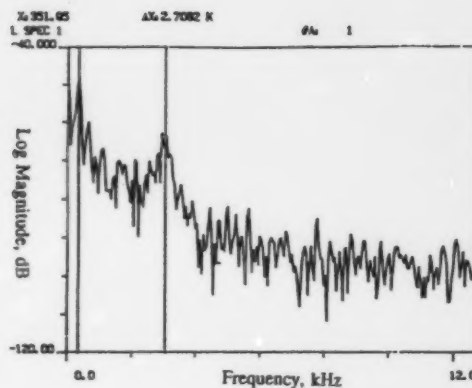


Fig.7. Spectrum of the intensity modulated signal with the fiber taper as a secondary strut at average flow velocity of 1.31 m/s and 0.633 μm light wavelength.

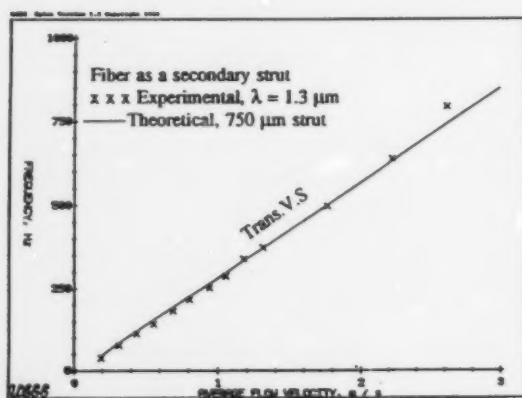


Fig.6. Detected frequency peaks with the fiber as a secondary strut, past a rigid shedding strut of 750 μm diameter, at light wavelength of 1.3 μm .

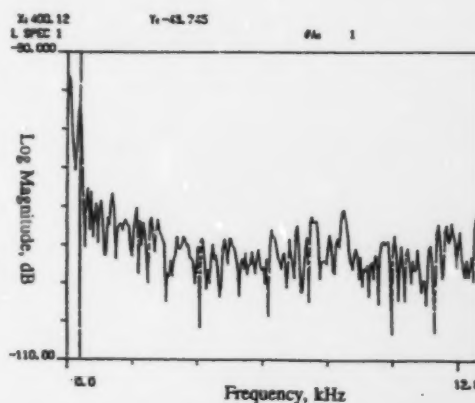


Fig.8. Spectrum of the intensity modulated signal with the fiber taper as a secondary strut at average flow velocity of 1.31 m/s and 1.3 μm light wavelength.

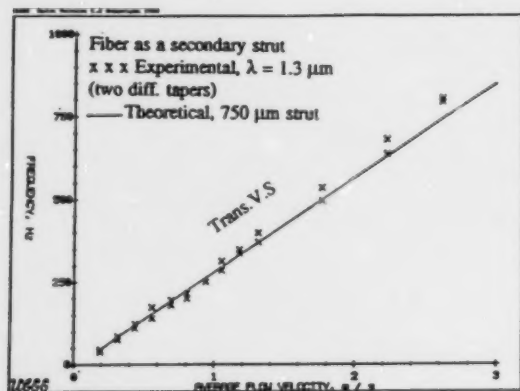


Fig.9. Detected frequency peaks with the fiber as a secondary strut past a rigid shedding strut of 750 μm diameter for two different fiber tapers of 23 μm waist diameter, 0.98 mm waist length, and 9.15 mm and 4.7 mm to the taper ends, and 29 μm waist diameter, 3.48 mm waist length, and 5.7 mm and 5 mm to the taper ends, at a light wavelength of 1.3 μm .

AN EXPERIMENTAL STUDY OF FREE CONVECTIVE FROST GROWTH ON A HORIZONTAL PIPE

P. H. Oosthuizen¹, G. Oram¹ and P.F. Monaghan²

¹ HEAT TRANSFER LABORATORY
Department of Mechanical Engineering
Queen's University
Kingston, Ontario, Canada, K7L 3N6

² Department of Mechanical Engineering,
University College,
Galway, Republic of Ireland

Frost growth rates on a cooled horizontal cylinder in the presence of a free convective flow have been experimentally measured and the effect of step changes in wall temperature have been examined. The apparatus used consisted of a long horizontal copper pipe through which a glycol-water mixture from a constant temperature bath was circulated. This pipe was mounted on a beam type arrangement and strain gauges on this beam were used to continuously determine the frost mass on the pipe. Tests were carried out with this pipe mounted in still indoor ambient air. The frost thickness on the tube was directly measured at various times using callipers. The variation of frost mass and thickness with time for various wall temperatures and various changes in wall temperature were measured. An equation for the mass transfer rate to the surface of the frost has been derived from the results.

NOMENCLATURE

b	=	Frost thickness (mm)
C	=	Constant
Gr	=	Grashof Number
K	=	Constant (mm/hr ²)
Sh	=	Sherwood Number
t	=	Time (hr)
T _f	=	Frost surface temperature (C)
T _{frz}	=	Freezing temperature of water (C)
T _{tub}	=	Surface temperature of frosting tube (C)

INTRODUCTION

Interest in the use of heat pump systems that have no forced flow over the external heat exchanger has recently revived. While in most such applications the wind will provide some measure of forced flow, in the worst case the heat transfer will, in such an installation, be by natural convection. Under some operational conditions frost will form on the heat exchanger, and therefore in order to model the performance of the heat exchanger under these conditions, a knowledge of frost growth rates with free convective flow is required. However, relatively little information on this situation is available. Frost growth rates on a cooled horizontal cylinder have therefore been measured in the present study. In an actual

heat pump installation, the heat exchanger temperature will change with time; therefore, in the present study, the effect of step changes in wall temperature have also been examined.

Frost grows on a cold surface in moist air when the temperature of the surface is below both the freezing point of water and the dew point of the air. Frost is composed of a mixture of ice and air. Its properties are strongly dependent on the way in which the frost was formed and on the age of the frost.

There have been several studies of frost growth in the presence of a natural convective flow; typical of these are those described in Refs. 1 to 9. None of these deal with single horizontal cylinders or with the effect of changes in wall temperature on the frost characteristics, i.e., with the subject of the present study. There have also been a number of studies of the growth of frost in the presence of forced convection (References 10 to 14). Some studies of frost growth in a natural outdoor environment have also been undertaken (References 15 and 16). Some moderate success has also been achieved with attempts to model frost growth under a particular set of circumstances (References 17 to 20).

APPARATUS AND PROCEDURE

The basic layout of the apparatus used in

the present study is shown in Figure 1. An ethylene glycol-water antifreeze mixture kept at a specified temperature in a constant temperature bath was circulated at a fixed, measured rate to a feed reservoir, through the horizontal 25 mm outside diameter copper tube on which the frost grew, to a drain reservoir and then back to the constant temperature bath.

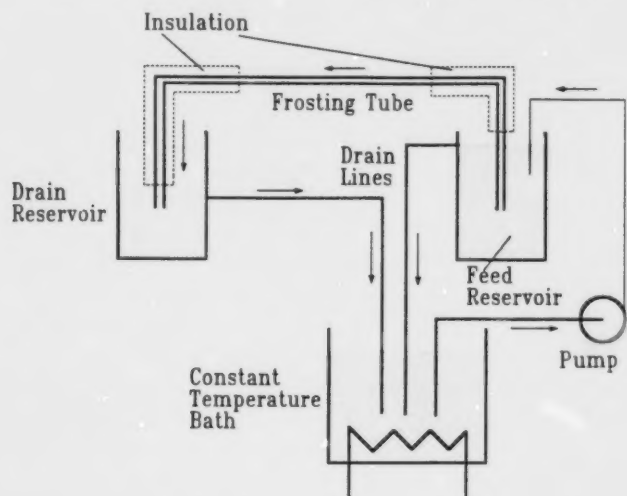


Figure 1 Overall arrangement of apparatus.

The frost mass was determined by measuring the load on a beam supporting the frosting tube. Because the total frost mass was expected to remain relatively it was necessary to eliminate the effects of the stiffness, which is temperature dependent, of any antifreeze supply tubes on the mass measurement. For this reason, the flow through the frosting tube was induced by a syphoning action as indicated in Figure 1, with the ends of the frosting tube sitting freely in the supply and drain reservoirs. An overflow mechanism was used to ensure that the fluid levels in the feed and drain reservoirs did not vary during a test as this would change the buoyancy force acting on the tube and thus interfere with the frost mass measurement. The frosting tube was mounted on a frame that was balanced on knife edges, and the weight of the tube was balanced by a counterweight (Figure 2). The unbalanced force produced by the frost growth was measured by connecting the frame to a cantilevered beam on which a strain gauge system was mounted. The strain gauge output was calibrated by hanging known weights from the frosting tube.

Thermistors were used to measure the temperature of the antifreeze at the inlet and exit of the frosting tube. The relative humidity of the ambient air was measured using a wet and dry bulb psychrometer through which air was being drawn by a small electrically driven fan. The ambient air temperature (the dry bulb temperature) was also given by this device. The tests were conducted with the apparatus mounted in an open laboratory, with no special precautions being taken to control the ambient temperature or humidity.

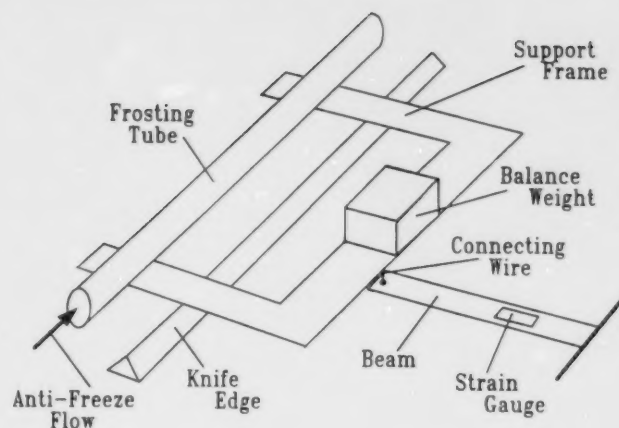


Figure 2 Arrangement of frost mass measuring system.

The apparatus was designed to measure the frost growth on the cooled frosting tube. The following were therefore measured at frequent regular intervals throughout a test which lasted typically for at least 50 hours:

- (1) Frost mass;
- (2) Inlet and outlet temperatures of the anti-freeze;
- (3) Ambient air temperature;
- (4) Ambient relative humidity; and
- (5) Time.

These measurements were obtained using an automatic data acquisition system. By using the known antifreeze flow rate through the frosting tube to determine the Reynolds number, it was determined that for its length-to-diameter ratio the flow inside the frosting tube was laminar and fully-developed. The Nusselt number for the flow inside the frosting tube could therefore be found. Furthermore, using the known anti-freeze flow rate and measured temperature rise across the tube, the heat transfer rate to the anti-freeze could be determined. Using these results, the temperature difference between the tube wall and the anti-freeze could be found allowing the mean tube wall temperature to be found at each time that measurements were taken. It was also deduced from the heat transfer rates that the temperature difference across the tube wall was negligible.

In addition to the measurements obtained using the data acquisition system, measurements of the frost thickness at the bottom, sides and top of the horizontal frosting tube were manually made using a pair of callipers at certain times during a test.

As mentioned in the previous section, one of the purposes of the present study was to determine the effect of fairly rapid changes in the tube wall temperature on the frost growth. In a given test, then, frost would be grown on the tube with the antifreeze temperature kept constant at a certain value. The antifreeze temperature would then be changed to a new value and the test continued, with the effect of the temperature change on the frost characteristics being studied. The antifreeze temperature, and hence tube wall temperature,

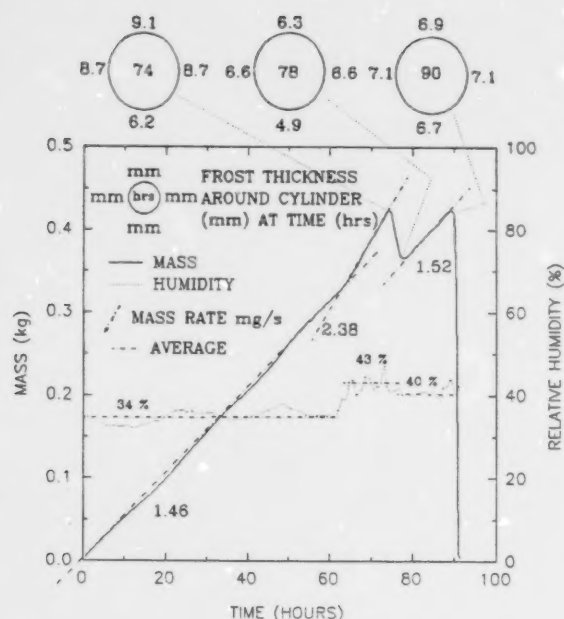


Figure 3 Variation of frost mass, relative humidity of air and frost layer thickness with time for test number 1. The line marked "mass rate" indicates the mean rate of mass accumulation over the period shown.

did not, of course, change instantly due to the thermal mass of the system. However, the time over which the temperature change occurred was relatively compared the total time of the test, e.g., the temperature change typically occurred over at most a 2 hour period while a test typically was at least 50 hours long.

More details of the apparatus and the test procedure are given in Reference 21.

RESULTS

Results obtained in five separate tests will be discussed here. Many other results were obtained but they all displayed the same basic characteristics as those discussed here. The frosting tube wall temperatures and ambient air temperatures during these tests are given in Table 1. It will be seen from this table that the changes in ambient air temperature during a given test were relatively .

The variations of frost mass and ambient relative humidity with time for three of the tests are shown in Figures 3 to 5. These Figures also show the measured frost thicknesses around the tube as measured at certain times during the tests. The tube wall temperatures during the tests are indicated in Table 1.

Figure 3 shows results for a very long duration test with a large rise in wall temperature late in the test. Because of the long initial period, the thickness of the frost on the tube is relatively large at the time of temperature change. As a result, when the wall temperature increases, the surface temperature of the thick frost layer increases until it

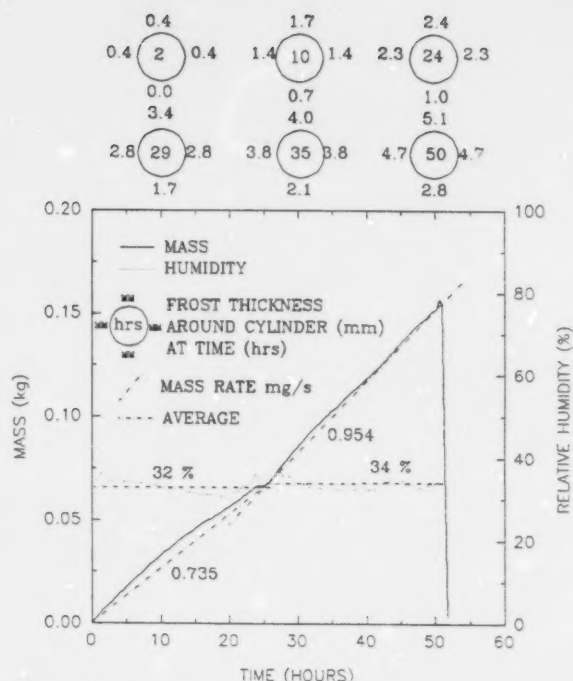


Figure 4 Variation of frost mass, relative humidity of air and frost layer thickness with time for test number 2. The line marked "mass rate" indicates the mean rate of mass accumulation over the period shown.

reaches the freezing temperature of water. Melt-back then occurs, the melting and dripping of condensed water from the tube was visually observed. During this period of meltback, the frost mass decreases. However, after this adjustment period the surface temperature of the frost again falls below the freezing temperature and the frost mass again starts to increase. It will be noted from Figure 3 that changes in the relative humidity occurred near the time that the antifreeze temperature was changed. The same changes will be seen to occur in all the tests. These changes in relative humidity were the result of the opening of the door to the laboratory in which the test was being conducted to allow personnel to enter the room to change the antifreeze temperature. It will be seen from the frost thickness measurements that, except after melt-back, the frost is thickest at the top of the tube and thinnest at the bottom. The frost layer was, in fact, always observed to be approximately flat across the bottom. The decrease in frost thickness associated with meltback can also be noted. After melt-back, the frost will be seen to be thicker at the sides than at the top due to the flow of melted water.

Figure 4 shows results for a case where the antifreeze temperature is decreased near the middle of the test (see Table 1). In this case the change in wall temperature has a very effect on the rate of growth of the frost mass. Frost thicknesses measured at a number of times during the test are also shown in this figure.

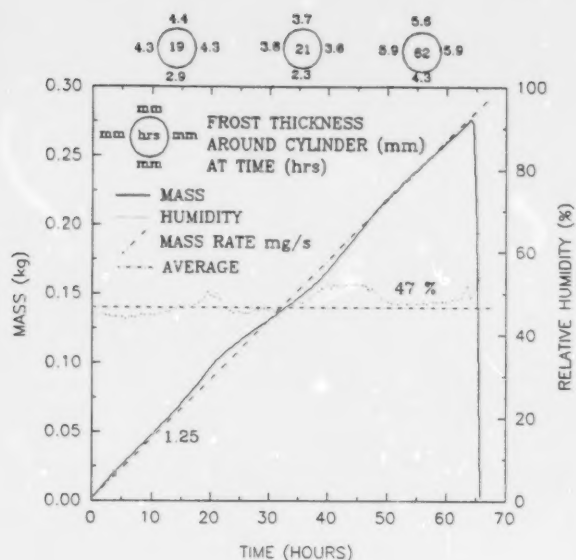


Figure 5 Variation of frost mass, relative humidity of air and frost layer thickness with time for test number 4. The line marked "mass rate" indicates the mean rate of mass accumulation over the period shown.

It will be seen that the thicknesses of the frost at the top, sides and bottom of the tube remain in roughly the same proportion to each other.

Figure 5 shows results for a case where the antifreeze temperature was increases relatively early in the test and by a relatively amount. A amount of melt-back was visually observed to occur in this case, but this had a very small effect on the frost mass, which grew at approximately the same rate throughout the test.

The results given in Figure 5 seem to indicate that, except when very significant amounts of meltback are occurring, the rate of growth of the frost mass is dominantly dependent on the relative humidity.

The variation in frost thickness on the side of the cylinder with time for all tests is shown in Figure 6. It will be seen from this figure that the frost thickness is approximately given on the average by:

$$b = K t^{0.5} \quad (1)$$

where the coefficient K depends on the temperature difference across the frost. The mean value of K derived from the present results was approximately $0.91 \text{ mm/hr}^{0.5}$.

The mean density of the frost was determined from the measurements of frost mass and thickness, and its variation with time for all tests is shown in Figure 7. Raising the antifreeze temperature, with resultant meltback, does produce some increase in frost density, while decreasing the antifreeze temperature tends to produce some decrease in density. The measured frost densities were almost all in the region 400 to 700 kg/m^3 .

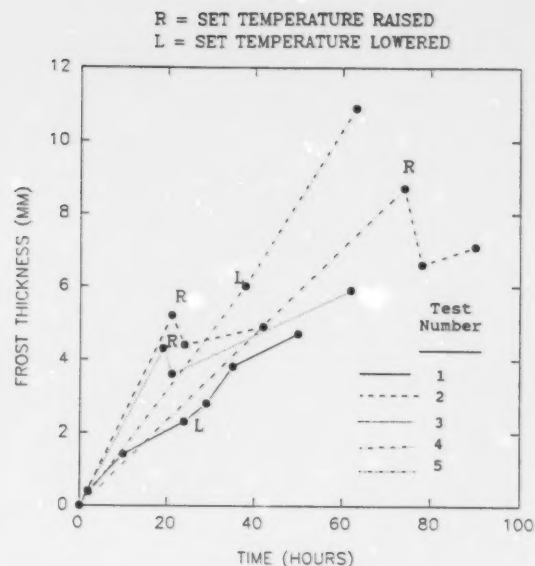


Figure 6 Variation of frost layer thickness on side of tube with time for all tests.

TABLE 1.

Test No.	Time Period - hours	Wall Temp. - °C	Air Temp. - °C
1	0 - 73	-9	26
	75 - 88	-2	27
2	0 - 24	-3	24
	25 - 50	-5	24
3	0 - 23	-4	23
	25 - 44	-1	24
4	0 - 19	-5	20
	21 - 63	-1	22
5	0 - 39	-1	24
	41 - 48	-5	24

The rate of increase in frost mass is dependent on the average rate of mass transfer to the outer surface of the frost. For the range of Grashof numbers covered by the present tests and because the buoyancy forces due to concentration differences can be shown to be quite negligible compared to those due to temperature differences, this is expected to be approximately given by an equation of the form:

$$Sh = C Gr^{0.25} \quad (2)$$

The Grashof numbers based on the difference between ambient air temperature and the estimated frost surface temperature and on the average outside diameter of the frost layer were all between 13,000 and 240,000.

In order to determine the value of the Sherwood Number, Sh, the mass transfer rate was found from the rate of growth of the frost, while the moisture concentration differences

were deduced using the measured ambient conditions and assuming that immediately adjacent to the surface the air is saturated and at the frost surface temperature. The frost surface temperature was not measured. However, the lowest value it could have would be the frosting tube wall temperature while the highest value it could have would be the freezing temperature of water. It has here rather arbitrarily been set equal to the average of these two values, i.e., it has been assumed that:

$$T_f = 0.5 (T_{frz} + T_{tub}) \quad (3)$$

Because the tube wall temperatures used in the present work were not very much less than the freezing temperature and because the ambient temperatures used were relatively high, the potential errors introduced by the this assumption are .

Using the values of the Sherwood number deduced in the manner discussed above, the value of the constant C in Eq. (3) was found to be 0.28, i.e., the mass transfer rate was found to be given by:

$$Sh = 0.28 Gr^{0.25} \quad (4)$$

This equation, together with Eq. (3) can be used to predict the growth in frost mass when melt-back is not occurring to an accuracy of better than 5% for conditions similar to those covered in the present study. It is perhaps worth noting that this equation gives mass transfer rates that are much lower than would be predicted using the Lewis Analogy. A similar result was obtained in Reference 6 for free convective frosting on a vertical flat plate.

CONCLUSIONS

Some of the main conclusions that can be drawn from the present study are:

(1) The biggest single factor effecting the rate of frost growth is the moisture concentration in the ambient air, which depends on the ambient relative humidity and temperature.

(2) The frost thickness was not uniform about the tube. When melt-back had not occurred, it was thickest at the top of the tube and thinnest at the bottom of the tube. The frost layer was almost flat across the bottom of the tube.

(3) Melt-back was observed to occur in all cases where the wall temperature was increased. Its effect was of significance, however, only were relatively large.

(4) The following equation:

$$Sh = 0.28 Gr^{0.25}$$

describes the experimentally observed mass transfer rates to an accuracy of within about 5%.

ACKNOWLEDGEMENTS

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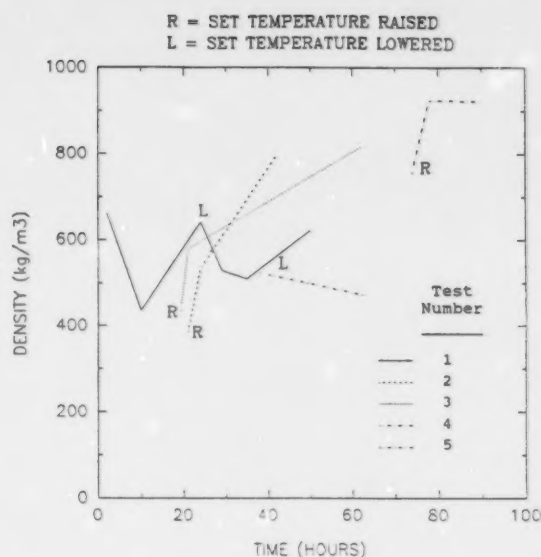


Figure 7 Variation of mean frost layer density with time all tests.

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WAVE PUMP POWER GENERATION

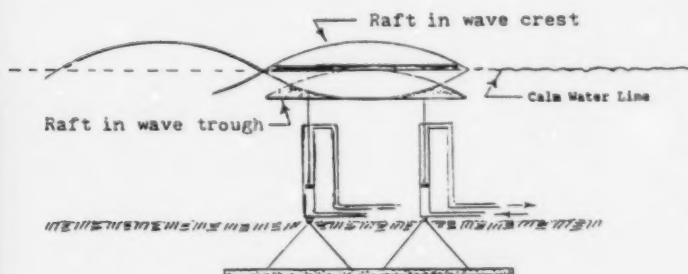
Ralph D. Wright, P.Eng.

The objective of this paper is to produce the preliminary design of a device which will be capable of extracting at least 75% of the total energy available in marine waves during high tide and low tide and during periods of high amplitude and low amplitude of wave action.

The diagram below shows the pump and raft arrangement which would harvest all of the wave energy for the type of wave shown. The raft begins its cycle in the leading trough of a wave. As the wave moves to the right the buoyant force increases until optimum pressure develops in the fluid discharge line and discharge nozzle opens allowing the high pressure fluid to drive turbine. The raft will continue to rise until the top of the raft coincides with the top of the wave crest. At this point in time there is no longer any water above the calm water line, only the space occupied by the hollow raft.

The paper deals with adjusting the size and shape of the raft plus the options of downstream equipment to conform with the requirements in the real marine environment. Due to variable wave lengths, (crest to crest distances) the raft length in this study is reduced to accommodate short waves. The power yield is therefore only 77% of full wave power potential.

A novel pump design which utilizes a roller chain crank shaft has been included. This pump facilitates the storing of water in elevated reservoirs for use in producing power when direct wave power is not sufficient for peak demands.



The selection of a project that would be pertinent to the international community should consider:

1. what portion of the world's population would benefit?
2. what would be the impact on their quality of life, resource depletion and self-sufficiency?
3. what would be the environmental effect, and
4. would the benefits go to those who need it most?

The establishment of clean energy generation in the trade winds belt would improve the quality of life in an area where alternate energy sources are not widely available, without damage to the environment, depletion of resources or dependency on outside imports.

For these reasons it is evident that marine wave energy merits close examination.

This paper is intended to analyze the concept of harnessing the energy of marine waves by using the vertical buoyant force of ocean swells acting on rafts attached to reciprocating pumps which in turn are anchored to the sea floor.

This wave pump system has the unique feature of providing nozzles at the discharge and which remain closed until the rising swell produces an

optimum buoyant force on the stationary raft. When this optimum force is reached, the nozzles open and the raft begins to rise.

There is a substantial amount of good news concerning wave energy conversion.

1. Waves gather energy from winds which are hundreds of miles away and transport it to shore lines in a concentrated accumulation of energy.

2. Due to the influence of the sea floor near the shore line the waves organize themselves in lines parallel to the shore line and advance like trained soldiers at convenient time intervals to deliver their energy to whoever has the ability to harness it.

3. Wave energy arrives in a heavy medium, namely water, which is much easier to harness than air. It is cheaper to build a raft from rough plank than to make the multitude of finely polished, shaped and balanced propellers that would be required to produce an equivalent amount of wind power.

4. Wave power can be gathered by pipe lines and delivered to a single generating plant.

5. Wave power can be stored in elevated water storage ponds and then utilized during peak load periods.

6. Wave power can also be used as a break water to provide calm beaches and improve harbour conditions.

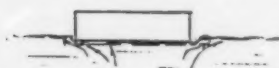
7. Wave energy can be produced in sufficient quantity and quality to provide the total electric power demand for countries with persistent on-shore winds, i.e. from Brazil to Cuba. The west coast on the south end of Chile is also probably worth considering.

To calculate the power that can be extracted from a wave using float powered pumps, the formula $BLK = P$ is suggested. B is the buoyant portion of the wave height used to provide pulling power on the pump. L is the remaining lift that is available for a specific height of wave. K is the length of the raft x the density of the water. For a raft 100 feet long, operating in salt water K becomes $100 \times 64 = 6400$. To obtain maximum power B is equal to L. For a wave height of 100 feet and a significant period of 10 seconds the gross power yield would be $5 \times 5 \times 6400 = 160,000$ ft. lbs. every 10 seconds or 16000 ft. lbs. per second. This amounts to 29 HP per ft. width of the raft. Losses due to pump friction, nozzles, pelton wheel and generator would reduce the power yield by about 30% which results in 20 HP or 15 KW per ft. width of raft.

It should be noted at this point that the power yield of a single raft operating alone would be less than what could be expected if the raft was operating beside other rafts.

When the water begins to rise in a trough which has many rafts close together, the rafts being restrained by the pumps forces the water to rise more rapidly between the rafts than it does on the outside face of the outside raft. The following diagram will illustrate this apparent gain in buoyancy development.

Single Raft



Multiple Rafts

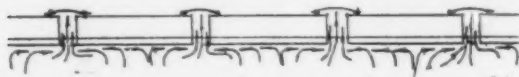


Fig. 1

The apparent advantage that a series of rafts placed close together have over a single raft is probably due to their ability to recover energy from the kinetic portion of the wave's energy.

I would estimate that placing the rafts close together would increase their power yield by approximately 50%. The net energy per ft. of raft width for single rafts would remain at 15 KW for single rafts and 22.5 KW for rafts close together when the waves are 10 ft. high.

Using 22.5 KW/ft. of raft width, a mile of rafts would produce over 100 megawatts of power when the waves are 10 ft. high and 50 megawatts of power when the waves are 7 ft. high.

Components of Wave Pump Power Generator

1. Floating rafts rectangular in shape. The length should be approximately 1/4 of the total crest to crest wave length that is common for ocean waves. (Fig. 2)

2. Reciprocal Pumps - with sucker rods attached to the rafts. The base of the pump cylinders are anchored by the ballast. (Fig. 2)

3. Ballast Assembly - The ballast is required to provide anchorage for the pump cylinders. Open top concrete containers are filled with rock and gravel. The submerged weight of the ballast should be double the buoyant capacity of the raft in order to remain stable when high seas lift the raft beyond the limit afforded by the pump cylinder. (Fig. 2)

4. The outflow Equalizer - ensures that the outflow waterline maintains a constant flow rather than the pulsating flow initiated by the piston pumps. (Fig. 2)

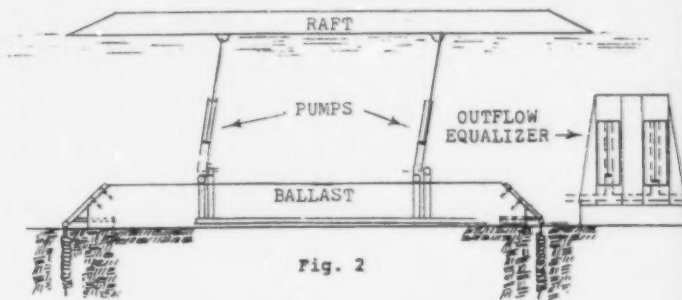


Fig. 2

Pump Details

5. The Inflow Header and Water Line brings non corrosive fresh water from the shore line and delivers it to each of the 10 pumps on each raft.

6. The Outflow Header and water line picks up pressurized water from each pump and delivers it to the on-shore pelton wheel power plant.

7. The Pump Neutralizer (Fig. 4) locks the valve on the pump open thereby neutralizing the pump. A check valve must be placed in the discharge at the top of each pump to prevent a

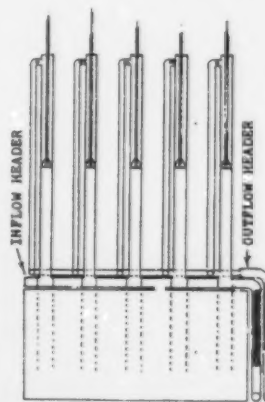


Fig. 3



Fig. 4

neutralized pump from short circuiting pressurized water. Neutralizing is required to accommodate optimum buoyancy pressure for periods of low wave amplitude. For 8 ft. waves use 8 of the 12" ID pumps to obtain 566 psi pressure at the optimum buoyancy of 1/2 of the wave height namely a 4 ft. For 4 ft. waves use 4 pumps and 2 ft. of buoyancy.

The pump neutralizer consists of a square bar linkage that passes through a keyway out in the sucker rod and is attached to the valve. A pipe

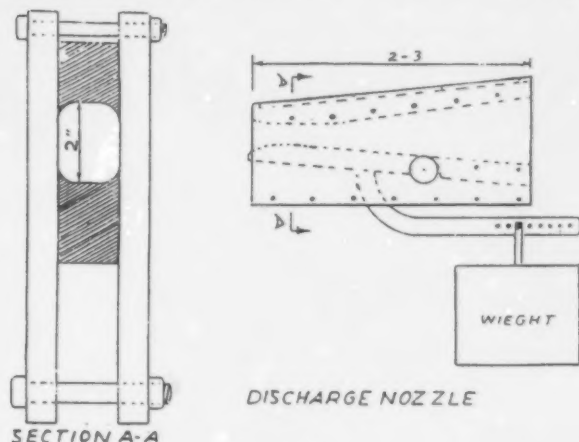


Fig. 5

acts as a shell to retain the square bar in the keyway. When the square bar is raised and locked, the valve is locked open. When the square bar is lowered, the valve moves up and down freely.

8. The Discharge Nozzles - are central to controlling operation of the pumps as well as maintaining a uniform velocity of jet flow to the cups on the pelton wheels. The nozzles remain closed until a predetermined pressure, say 600 psi is developed in the outflow line. This locks the raft down until optimum buoyancy is established by the water rising around the rafts. At this point the nozzles will open which unlocks the movement of the rafts, and the raft will rise upward in the power stroke delivering high pressure water through the nozzles to the pelton wheel. Should the raft rise slower than the water line, more buoyancy pressure is applied, say 660 psi. The nozzle responds by opening further, back pressure drops, and the rising of the rafts accelerates. If the rise of the rafts exceeds the rise of the water line, buoyancy and pressure is decreased and the nozzles throttle back slowing down the flow rate to the pelton wheel. (Fig. 5)

9. The Water Storage Reservoir - should be located near the power site if sufficient area can be found at preferably 1200 ft. of elevation above the power plant site. This would enable the plant operator to divert excess water @ 566 psi from the pelton wheel supply line to the elevated storage for back up power supply during peak loads or during periods of substandard wave activity.

Should there be no water storage site at the preferred elevation, a portion of the wave pump rafts could be equipped with larger pumps producing lower pressure and higher volume for supplying water to less elevated storage sites.

To accommodate a 24 hour supply for a group of 10 rafts powered by 10 ft. waves and delivering water to a 1200 ft. of elevation would require 63 acre ft. of storage. A 5 acre pond with an average depth of 12.6 ft. would be sufficient.

Off-Site Storage Reservoirs - can be located wherever there is a reasonable large water supply adjacent to an elevated storage site. In this case the power is transported to the remote reservoir site by high voltage electrical transmission lines. The electrical power is then available to provide power to pump the water up to the elevated storage to accommodate peak load power generation when required. The net loss would be the loss in the electric motor, the second generation of power, and the pump losses which would be 15%, 15% and 5% respectively.

The Reservoir Supply Line - is also used as the reservoir outflow line required to power the pelton wheel generation unit.

10. The Reservoir Generator - is an induction in reverse rotation. It is used as a motor to power the pump when filling the reservoir.

11. The Shoreline Pelton Wheel - is designed to operate at a rim speed which is 45% of the speed of the water jet from the nozzle. The jet speed of 277 ft. per second results from converting the 566 psi supply pressure to static head and then to velocity head.

$$\text{Static head of } 566 \text{ psi} \times \frac{144}{64} = 1273.5 \text{ ft.}$$

Allow 73.5 ft. for pipeline losses = 1200 ft. static head

$$\frac{V^2}{2g} = 1200 \text{ ft.} \quad V = 277 \text{ ft./sec}$$

Pelton rim speed = 1/2 of 277 = 138 ft./sec. or 8280 ft./min.

Pelton wheel diam. for RPM of

$$600 = \frac{8280}{600\pi} = 4.4 \text{ ft. diam.}$$

12. The Lock Down Valve - is required to protect the rafts from the ravages of a violent storm. When the height of wave exceeds the generating limit of the generating system there will be an added force to the normal buoyancy force due to the excessive kinetic energy from the rapid movement of the wave. This will result in a substantial increase in pressure in the delivery line. A motorized valve in the delivery line will respond to the pressure surge and will close the delivery line. This will prevent any

upward motion of the rafts. The valves in the pump pistons continue to open on the down stroke which enables the raft to move downward, however the raft cannot move upward as long as the delivery line is closed. The rafts remain jacked down below the turbulent upper water. When the storm subsides, the Lock Down valve is manually opened very slowly until all the rafts rise to the surface ready to continue operation.

13. The Raft Coupler - is a flexible belt running along the length of the top edges of two adjacent rafts. The belt is perforated to allow air to escape, however the holes are too small to allow a significant amount of water to pass through when the rapid upward surge of water beings to rise between the rafts with the advance of the wave crest. When the water between the rafts becomes static, and exerts a back pressure on the water under the raft, the raft becomes fully buoyant. The Raft Coupler also has a chain linkage connecting the rafts opposite the sucker rod connection both fore and aft. Used rubber tires can be added to prevent the rafts from coming too close together.

14. The Reservoir Supply Pump - is a double acting piston pump which is required to provide the high lift which is desirable for economical power storage. The higher the lift, the smaller the quantity of water required. As a result, the size of reservoir and reservoir supply line is reduced. The Reservoir Supply Pump consists of two cylinders with two pistons in each cylinder. (Fig. 6)

The top piston Item 1 is powered by sucker pipe Item 2. The lower piston Item 3 is powered by sucker rod Item 4 which passes through the sucker pipe extending beyond the top of the sucker pipe and is secured to the upper slider of a pump jack. The sucker pipe passes through the top seal of the pump cylinder and is attached to the lower slider of the pump jack. The pistons must move at the same rate and in opposite directions at all times.

I have built and tested a model of this pump using foot operated treadles and a synchronizing linkage between the sucker pipe and the sucker rod. The operation was very smooth with high volume output.

15. The Pump Jack - is driven from the bottom by roller chains Items 16 (Fig. 7) which drives the lower roller chain crank shaft Item 14. The sucker pipe Item 2 is attached to the lower slider Item 19 which in turn is attached to a roller chain Item 14 by a heavy pin connection Item 18. As the roller chain moves upward the slider and sucker pipe move upward. When the pin connection Item 18 rounds the top sprocket it is moved to the opposite side of the slider and pulls the slider downward as it moves through the down side of the cycle. The sucker rod Item 4 is attached to the upper slider which is driven by the upper roller

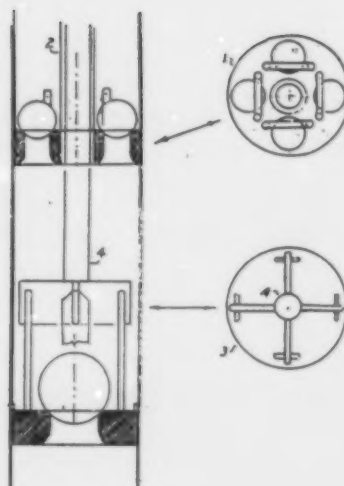


Fig. 6

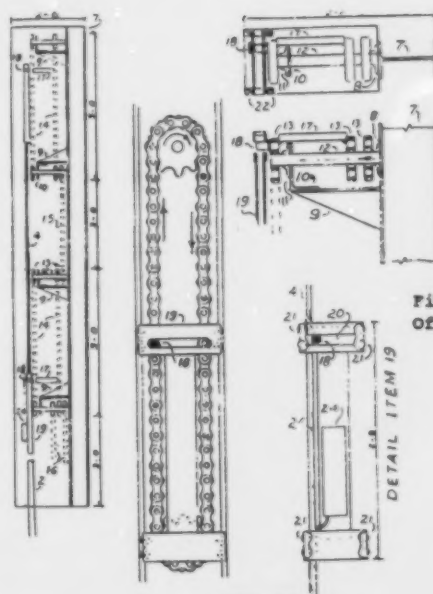


Fig. 7
Offset Drive

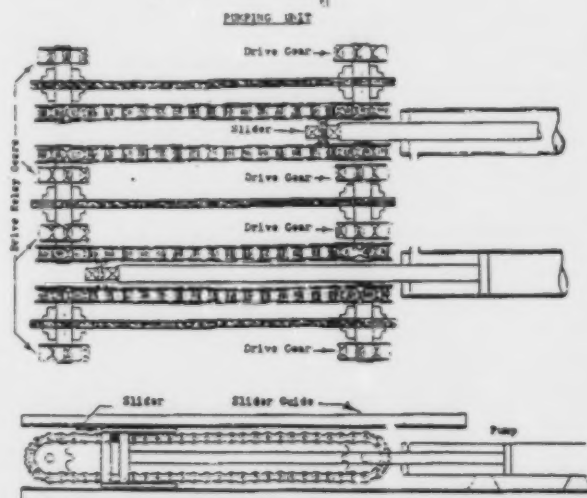


Fig. 7a - Split Drive

chain crank shaft which in turn is powered by the timing chain Item 15 which also synchronizes the movement of the sliders to ensure that they move in opposite directions. (Fig. 7)

Fig. 7a shows a split drive system in lieu of offset drive shown in Fig. 7.

The advantage of using the roller chain crankshaft instead of a rotary crankshaft is due to the uniform rate of travel of the piston except for a brief transition during change of direction. A stroke of 6-0 can be achieved by attaching the slider to a roller chain drive which passes over sprockets spaced 6-0 apart. A conventional pump with a 3-0 stroke would require two complete cycles to achieve the same pump displacement. However when the same roller chain stress and the same maximum piston velocity are applied to both pumps, the total outputs are radically different as is shown graphically in (Fig. 8).

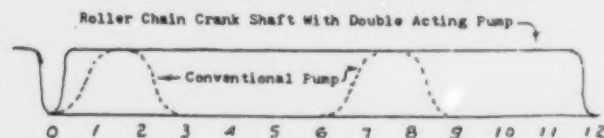


Fig. 8

The powering of reciprocal motion by the rotary drive from an electrical motor using the roller chain crank shaft can also be reversed. The reciprocal motion of a steam drive piston can be converted by the roller chain crank shaft to the rotary motion required in the drives of tractors, bush buggies or river boats.

16. Off-Shore Power Generation can be considered by using electric cable to transmit power generated at the raft location. The power could be generated on the sea floor (Fig. 12) or on board the raft (Fig. 9).

The rafts would power hydraulic pumps to produce high pressure oil which in turn is used to drive a hydraulic motor coupled to a generator. The pumps can be individually neutralized as with the on-shore system by opening the bypass valve A which would allow the oil to bypass the motor. The air tank would have to be charged with compressed air to ensure a reasonable consistent flow of oil through the motor.

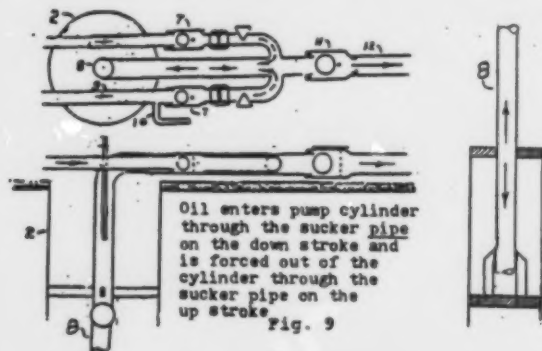
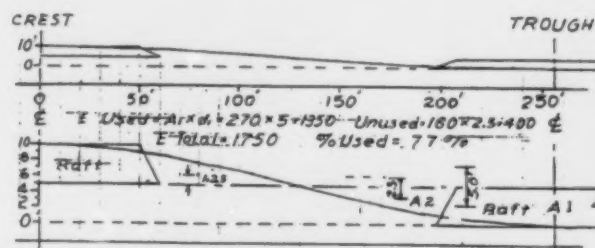


Fig. 9

The Trochoidal Curve - shown below shows a profile from the centre of the crest to the centre of the trough. The upper curve shows the true shape of a 10 ft. wave 512 ft. in total length. The lower curve shows the same curve with the vertical scale enlarged four times.

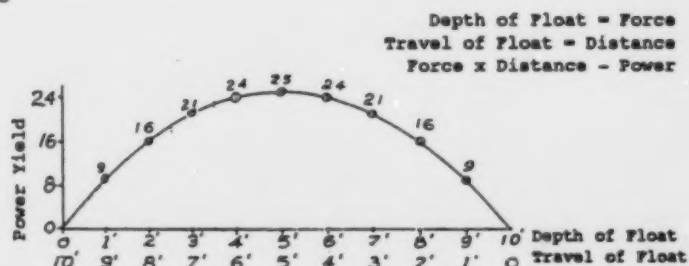
It should be noted that the curve in its true shape is very flat at the crest and at the trough. Although the power yield could be increased by increasing the length of the raft there would be diminishing returns from the increase in construction costs.

It should also be noted that when a wave reaches shallow water nearer to the shoreline, the length of wave will decrease and the height will increase. For this reason the length of the raft has been shortened to 1/4 the length of a 400 ft. wave rather than using 1/4 the deep sea wave length of 512 ft. as shown in the plotted curves. (Fig. 10)



Trochoidal Curve
Fig. 10

The Power Curve - shown below shows how the buoyancy portion of total wave height relates to the length of power stroke and also to the power yield. It is evident that changing the buoyancy factor by plus or minus 20% from the optimum 5 ft. for a 10 ft. wave only decreases the power yield by 4%. (Fig. 11)



POWER YIELD CURVE - (for 10' wave)
Fig. 11

NOTES: The total cost per raft installed, 3000 ft. offshore, and including Pelton wheel and generation equipment plus costs for transportation, administration, insurance and supervision is estimated at \$280,000.00 per raft. Should the wave power be low and two rafts are required to provide the designed capacity of generation, the second raft would add \$68,000.00 to the cost. No reservoir costs have been included.

The cost of \$280,000.00 for 450 KW of generation capacity is \$622.00 capital cost per KW of generation capacity. If the second raft is required, the capital cost would rise to \$773.00 per KW of generation.

The problem of flow equalization is largely eliminated when a long string of rafts are located on an alignment which is inclined at a small angle (15°) to the wave front. Rafts in small groupings can be set back in order to delay contact with the swell and therefore produce their power in between the power strokes of neighbouring rafts.

A small air tank connected to each water line could be used to cushion any hard shocks in the pipe line.

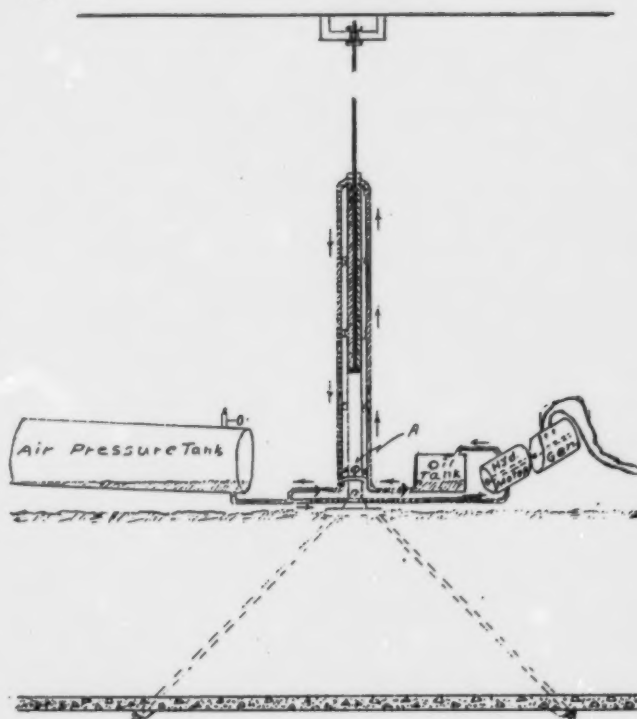
CONCLUSIONS: Wave power is available in great quantity providing key problems are solved and providing the term of reference demand that the full amount of recoverable energy is harnessed.

It should be understood that devices that pick up only a portion of recoverable energy are unacceptable because the amount of prime shoreline with strong year round onshore winds are limited. This very valuable resource should be respected. The ultimate objective should be to construct equipment which takes nearly all of the energy out of the waves and leaves only calm water behind.

Canada has exported electric power plant designs to various countries i.e. Romania, Korea and India. It would be to our advantage to develop a wider selection of power plant design that would reflect what is appropriate for the area which we are attempting to assist.

The most appropriate location for the design being proposed would be where there are strong onshore winds, deep water close to the shoreline, and high land elevations along the shoreline for reservoir location. The south coast of Chile may have sites that meet these requirements.

Fig. 12



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TECHNICAL SESSION 7
MATERIALS-DESIGN ANALYSIS II

Chair: R. Seshadri



FINITE ELEMENT MODELING OF TILLAGE TOOL DESIGN

R.L. Kushwaha, Professor
Department of Agricultural and Bioresource Engineering
University of Saskatchewan
Saskatoon, Sask. CANADA
S7N 0W0

ABSTRACT

A non-linear finite element model was developed for three dimensional soil cutting by tillage tools. A hyperbolic constitutive relation for soil was used in the model. Analysis was carried out to simulate soil cutting with rectangular flat and triangular tillage blades at different rake angles and with curved blades. Interface elements were used to model the adhesion and the friction between soil and blade surface. Soil forces obtained from the finite element model for the straight blades were verified with the results from laboratory tillage tests in the soil bin. The finite element model predicted draft force accurately for both tillage tools. Results indicated that the draft was a function of rake angle, tool shape and the curvature.

INTRODUCTION

A large amount of energy is consumed during tillage operations. Even though no-till practice has been introduced recently, a furrow opener is usually required to cut the soil for seed placement. The soil-tool interaction result in frictional and wear losses of tools causing excessive draft. Because of high cost of energy and significant losses due to the friction and wear, it is desirable to develop highly efficient tillage tools which will require less energy to provide a satisfactory soil environment for seed emergence.

Initially, a theoretical approach to the soil cutting problem was formulated based on the well-known Terzaghi's passive earth pressure theory (Terzaghi, 1943). Several three dimensional soil cutting models were proposed based on Terzaghi's passive earth pressure theory by different researchers (Hettiaratchi and Reece 1967, Godwin and Spoor

1977, McKyes and Ali 1977, Perumpral et al. 1983, and Swick and Perumpral 1988). These models usually provided fairly simple equations to evaluate the soil forces on the tool. However, most of the above models can only be used for a static situation. Because of the simplifications made during modelling, none of the above models could account for the effect of different tool shapes, especially if the tools were curved.

With the accessibility of computers, the numerical method was also developed to solve the soil cutting problem. Yong and Hanna (1977) first proposed a finite element model for two-dimensional soil failure with a wide blade. Chi and Kushwaha (1989) developed a three dimensional finite element model for a narrow cutting blade. This finite element model showed a capability of simulating different tool shapes. Using this model, analyses were conducted on flat and triangular blades and on curved blades (Chi and Kushwaha, 1990 & 1991).

This paper compares the draft requirements of straight rectangular, triangular and curved blades obtained from the three-dimensional finite element model. Results from the finite element model for the rectangular and triangular blades were verified with the results from soil bin tests.

PROCEDURE

Finite Element Model

A three-dimensional finite element mathematical model for predicting soil forces with simple tillage tools was developed by Chi and Kushwaha (1989). The finite element model utilized an incremental procedure to solve the

nonlinear behavior of soil and the interaction between soil and tool surface. The finite element model can be described by the following equation:

$$\int_{\Omega_e} (D\Phi)^T C D \Delta u_j^e d\Omega_e = \int_{\Omega_e} (D\Phi)^T \sigma_{oc} d\Omega_e + \int_{\Omega_e} \Phi^T f d\Omega_e - \int_{\Gamma_e} \Phi^T p d\Gamma_e \quad [1]$$

Where:

- D = differential operator matrix.
- Φ = shape function matrix.
- C = constitutive matrix.
- Δu_j^e = nodal displacement increment vector.
- σ_{oc} = initial stress vector.
- f = body force vector.
- p = surface compression vector.
- Ω_e = element domain.
- Γ_e = boundary surface of the element.

The above equation can be written in short terms as below:

$$K^e \Delta u_j^e = F^e \quad [2]$$

where K^e is element stiffness matrix and F^e is element load vector. The element load vector includes three terms: 1) initial stress term, 2) body force term, and 3) the surface compression term.

Constitutive Relationship of Soil

As the soil is a non-linear stress-strain material, the tangent modulus, (E_t), changes with the state of stress. Kondner and Zelasko (1963) proposed a hyperbolic model to represent a typical stress-strain relationship of the soil. Duncan and Chang (1970) developed an equation of the tangent modulus, (E_t), based upon Kondner and Zelasko's model, which is given below:

$$E_t = \left[1 - \frac{R_f (1 - \sin \phi) (\sigma_1 + \sigma_3)}{2 c \cos \phi + 2 (\sigma_1 + P_a) \sin \phi} \right]^2 K P_a \left[\frac{\sigma_3 + P_a}{P_a} \right]^n \quad [3]$$

Where:

- E_t = tangent modulus of soil.
- c = cohesion of soil
- ϕ = internal friction angle of soil
- P_a = atmospheric pressure
- σ_1 = major principal stress
- σ_3 = minor principal stress
- R_f = failure ratio
- K, n = dimensionless numbers

In this equation, the tangent modulus, (E_t), of soil was expressed as a function of major and minor principal stresses. The failure ratio, (R_f), in Eq. [3] was defined as the ratio of ultimate deviatoric stress to the soil strength. This tangent modulus, (E_t), and the Mohr-Coulomb failure criterion were utilized in this model. The constitutive matrix is given by the equation [4].

Triaxial tests were carried out to determine the soil parameters in Eq. [3] and Eq. [4] (Chi and Kushwaha, 1988) for the soil used in verification of the model. The parameters obtained from the test are listed in the Table I.

$$C = \frac{E_t}{(1+\nu)(1-2\nu)} \begin{bmatrix} 1-\nu & \nu & \nu & 0 & 0 & 0 \\ \nu & 1-\nu & \nu & 0 & 0 & 0 \\ \nu & \nu & 1-\nu & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{1-2\nu}{2} & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1-2\nu}{2} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{1-2\nu}{2} \end{bmatrix} \quad [4]$$

Where:

ν = Poisson's ratio

Table I. Parameters used in the finite element analysis.

Term	Value
Parameters in soil constitutive relationship:	
K	8.10
n	0.0
cohesion of soil (c)	7.19 kPa
internal friction of soil ϕ	34.5°
soil gravity γ	14.34 kN/m ³
poisson's ratio ν	0.329
failure ratio R_f	0.719
Parameters of soil-tool interface:	
K_i	265.66 kPa
n_i	0.8437
adhesion C_i	3.288 kPa
external friction δ	23.5°
failure ratio R_{fi}	0.89
Tillage blade parameter:	
rake angle α	30° - 90°
depth d	10 cm
width w	5 cm

The soil in the soil bin was a classified as clay loam (sand 47.5%, silt 24.2%, clay 28.3%). The average soil density and moisture content during tests were 14.34 kN/m³ and 14.8% (d. b.).

Interface element

Adhesion and friction characteristics of soil always react on the surface of cutting blade. These two soil characteristics significantly influence draft and vertical force. Thus, in order to account for the effect of these variables, interface elements were inserted between the soil and tool surface.

Several constitutive models have been developed for soil-tool interface. Desai et al. (1984) developed an elasto-plastic interface model, in which the shear stress increased linearly with the relative displacement before the shear stress reached its maximum value. After reaching its maximum

value the shear stress remained constant. The tangent shear modulus becomes zero after the maximum shear stress is reached. Bekker (1960) proposed an exponential stress-relative displacement function for loose soil. Clough and Duncan (1971) proposed a hyperbolic model for the interface between the soil and wall.

Laboratory tests were conducted in a modified shear box to study adhesion and friction characteristics between soil and the tillage tool metal. The shear stress and relative displacement curves obtained from shear box tests were largely nonlinear. The results also show that the hyperbolic model (Clough and Duncan, 1971) provided a better prediction of the test data.

An interface element was proposed by Chi and Kushwaha (1991). The tangent modulus of the interface element was required in the incremental procedure of finite element analysis. For the hyperbolic model the tangent modulus is derived from Eq. [4] as:

$$K_t = K_f \left(\frac{\sigma_n + P_a}{P_a} \right)^{n_1} \left(1 - \frac{R_{f1} \tau}{\tau_{max}} \right)^2 \quad [5]$$

where:

- K_t = tangent modulus of soil-tool interface
- R_{f1} = failure ratio of soil-tool interface
- τ_{max} = maximum shear stress
- τ = shear stress
- K_f, n_1 = parameters obtained from tests

The maximum shear stress was defined as:

$$\tau_{max} = C_a + \sigma_n \tan \delta \quad [6]$$

where:

- C_a = adhesion between soil and cutting blade
- σ_n = normal stress
- δ = friction angle between soil and blade

Soil shear tests with a soil shear box were conducted under different normal stresses. The parameters for the interface element are also listed in Table I.

Stress Correction

The tangent modulus in the constitutive matrix was calculated from the element stress. The element stress also contributed to the calculation of the reaction force on the tool. In the incremental procedure, the soil stress increases with the increase in the blade displacement. The soil stress should not fall outside the failure surface represented by Mohr-Coulomb model during loading. However, since a residual modulus was assigned to the element after the element failure, it was possible to have a stress level outside of the failure surface. This incorrect stress level could affect the boundary force calculated in the later increments. In this case, the stress level should be corrected back to the failure surface. Davison and Chen

(1974) showed a procedure to correct the stresses at constant mean stress for plane stress analysis. Nayak and Zienkiewicz (1972a) and Siriwardane (1980) proposed a procedure to correct the stresses normally to the yield surface.

Since three-dimensional stresses were involved in soil failure with a tillage tool, it was more convenient to define a normal direction to the failure surface. Thus, the stresses were corrected normal to the failure surface by using Mohr-Coulomb criterion. Nayak and Zienkiewicz (1972b) transferred the Mohr-Coulomb criterion into a more convenient form by using the stress invariants, as given by the following equations:

$$F = -\frac{I_1}{3} \sin \phi + \sqrt{J_2} \cos \phi_0 + \sqrt{\frac{J_2}{3}} \sin \phi_0 \sin \phi - c \cos \phi \quad [7]$$

and

$$\sin \phi_0 = -3 \frac{\sqrt{3}}{2} \frac{J_3}{(\sqrt{J_2})^3} \quad [8]$$

where:

- F = failure function of Mohr-Coulomb criterion
- c = cohesion of soil
- ϕ = angle of soil internal friction
- I_1 = first invariant stress tensor
- J_2 = second invariant of deviatoric stress tensor
- J_3 = third invariant of deviatoric stress tensor

A positive value of failure function (F) indicates a stress level outside of the failure surface. During the stress correction, the stress state is brought back to the failure surface in the normal direction. The corrected stress increment required to bring the stress level back to the failure surface is derived as:

$$(d\sigma) = \frac{-F(\sigma^1) \left(\frac{\partial F}{\partial \sigma} \right)}{\left\{ \frac{\partial F}{\partial \sigma} \right\}^T \left\{ \frac{\partial F}{\partial \sigma} \right\}} \quad [9]$$

where:

- σ^1 = stress level before the stress correction

The corrected stress level is given by the following equation:

$$\{\sigma\}_2 = \{\sigma\}_1 + (d\sigma) \quad [10]$$

where:

- $\{\sigma\}_2$ = stress after correction
- $\{\sigma\}_1$ = stress before correction

Since the gradient $\partial F / \partial \sigma$ was calculated based on the stress outside of the failure surface, it may not bring the corrected stress exactly back to the failure surface. Therefore, an iteration method was used and the procedure was repeated until a satisfactory convergence was reached.

Finite Element Mesh

The tetrahedral constant strain elements were used during the analysis because of their simplicity and convenience for analysis of nonlinear material. It was simpler to generate the finite element mesh for the narrow rectangular flat blade and for the narrow triangular blade. However, for the curved blade, the tool angle changed with depth. In order to simplify the problem, the circular curves were used for blade shape. The tool angle at soil surface was selected as vertical (90 degree rake angle) and the tool angle at bottom of the blade was changed as 90 degree (straight blade), 60 degree, 30 degree and 0 degree (most curved blade). Fig. 1 shows a typical finite element mesh for the curved blade used in the analysis. Similar meshes were used for the rectangular and the triangular blades.

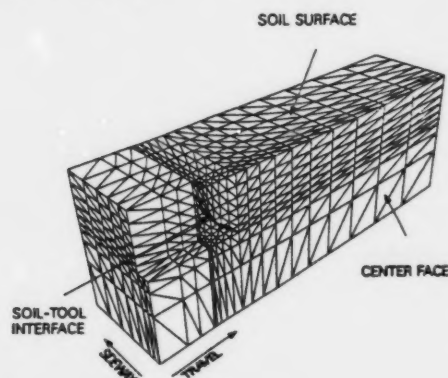


Fig. 1 A typical finite element mesh for the curved blade

In all cases, the soil displacement and failure was symmetric about the center line of the blade. Thus only half of the total region was considered in the analysis. The region of influence considered in the analysis had a length of six (vertical blade) to seven (most inclined tool) times of the tool operating depth (one tool operating depth behind the tool and five to six tool operating depths ahead of the tool), a depth of 2.5 times of the tool operating depth and a width of six times of the half tool width. Because of the symmetry, no side movement occurred on the center plane.

VERIFICATION OF THE MODEL

Soil Bin Test

The tillage laboratory tests were conducted to verify the results of the finite element model for the rectangular and the triangular blades. The soil preparation was carefully controlled during the test in order to provide a relatively uniform soil condition throughout all test runs.

In the soil preparation procedure, the proper amount of water was sprayed uniformly

onto the soil to obtain the desired soil moisture. Then the soil was covered with a plastic sheet to allow the moisture to infiltrate from the soil surface down to the soil below. The soil was root-tilled, levelled, and packed. Two different packers were used to pack the soil. The soil was first packed with a sheep-foot packer, which compacted the sub-surface soil and finally, a smooth packer was used to pack the surface soil.

Five core samples were taken from different location of the soil bin for soil density and moisture content measurement. The average soil density and soil moisture were used in the triaxial tests and shear box tests to determine the soil parameters.

The rectangular flat blade (Fig. 2) and the triangular blade (Fig. 3) were tested in the soil bin. The working width of both blades was 50 mm. The triangular blade was made from an angle iron piece. Tillage forces were measured with a force transducer equipped with six load cells. One load cell measured draft, two measured vertical force and the other three measured the side force. The force signals were sampled by an HP-3701A data logger and then fed into HP1000 computer. The tests were conducted at a travel speed of 2 km/h. Four test runs were repeated for each soil preparation.

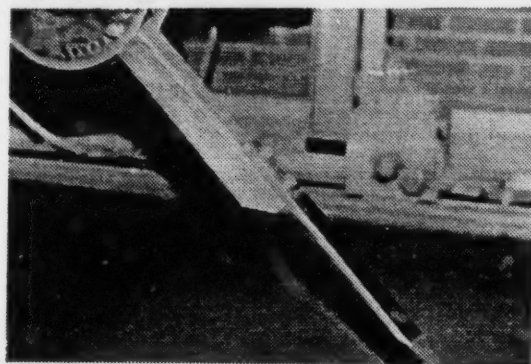


Fig. 2 Rectangular flat blade in the soil bin



Fig. 3 Triangular blade in the soil bin

RESULTS AND DISCUSSION

Force-displacement curves were obtained from the finite element model similar to the one shown in Fig. 4 for the rectangular flat blade. Displacement loading was used in the finite element analysis. The draft force increased as tool moved along the direction of travel. When the displacement increased to a certain point, the soil ahead of tillage tool collapsed. A maximum draft force was obtained at this failure point (Fig. 4).

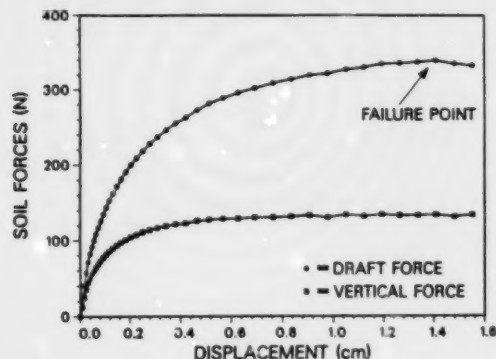


Fig. 4 Force-displacement curve for the rectangular blade at 45° rake angle

The draft and vertical force from the finite element model were verified with the results from soil bin tests as shown in Fig. 5 and Fig. 6. The comparison of draft forces shows that the finite element model predicted relatively accurate draft force. The direction of the vertical force changed with different rake angles (Fig. 6). A negative vertical force indicated a force tended to lift the tillage tool upwards. For the inclined tool (both rectangular flat and triangular blades), the finite element model slightly over-predicted the vertical forces. However, for the vertical tool, the finite element model under-predicted the vertical forces due to a slight bending of the blade during the tillage which would make the actual rake angle greater than 90-degree. For the finite element model, the blade was idealized as a rigid body.

The draft is the most important force which is directly related to the energy used in a tillage operation. Both the finite element model and the soil bin tests showed that the draft was reduced by using a triangular blade instead of the rectangular flat blade for large rake angles, especially in the vertical position. However, for a rake angle less than 45 degrees, there was no significant difference in drafts between the two blade shapes. The results also showed that the draft force decreased as rake angle decreased, but remained almost constant below the rake angle of 45-degree.

The predicted draft for different curvatures of the curved blade are shown in Fig. 7. For a 90-degree tool angle at soil

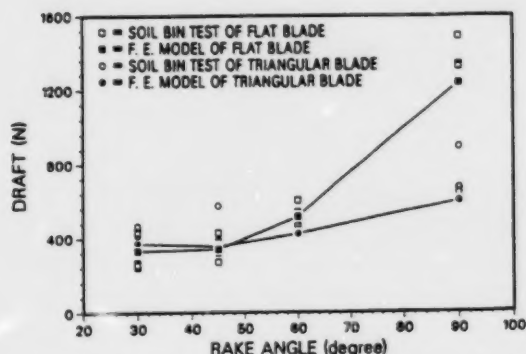


Fig. 5 Draft of the rectangular flat and triangular blades.

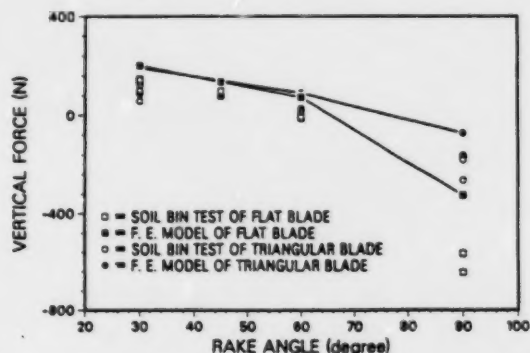


Fig. 6 Vertical force of the rectangular flat and triangular blades.

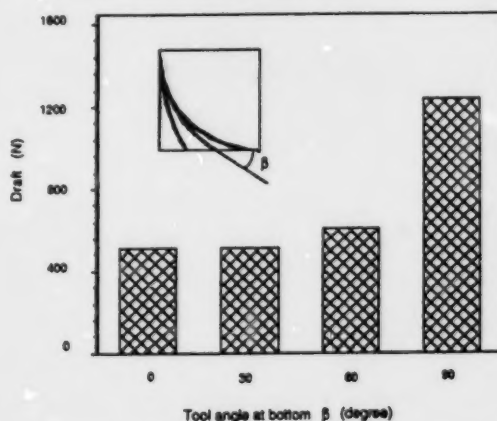


Fig. 7 Draft of a curved blade at various tool angles

surface, increasing curvature decreased the angle at bottom end of the soil-tool interface. The resistance to soil movement was reduced and lower draft values were obtained at the smaller angles (Fig. 7). The minimum draft was obtained at zero and 30-degree tool angle at bottom.

Figures 5 and 7 indicate that the draft of the three blade shapes was nearly equal at the rake angle of 30 degrees. The face of the

curved blade was also rectangular which would have resulted in the similar values as for the flat bale. However, the procedure indicates that several tool shapes could be examined using the finite element model to determine a suitable shape, tool curvature and rake angle which would result in the least energy requirement.

CONCLUSIONS

A three-dimensional finite element model was developed to simulate soil cutting with a tillage tool. The model could be used to determine the optimum shape and design parameters of a tillage tool which would require the least energy.

Laboratory tillage tests conducted in the soil bin to verify the results from the finite element model showed that the finite element model provided reasonably accurate prediction of the draft forces for both rectangular flat and triangular blades. The finite element model also showed a good prediction of vertical force for the inclined blade.

Both the finite element model and soil bin tests showed that the triangular blade required less draft for large rake angles, especially for the vertical position. However, for a rake angles less than 45 degrees, there was no significant difference in drafts between the two blade shapes.

For the curved blade, the draft increased with the increase in the curvature. The minimum draft was obtained for zero and 30-degree angle at the soil contact surface.

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THE CREEP ANALYSIS OF PRESSURE VESSEL COMPONENTS

W.A. MACFARLANE

SNC-INTEG, VANCOUVER, BRITISH COLUMBIA

In elastic analysis it is common to establish the separate membrane solutions for each component and then to superpose self equilibrating discontinuity loads which are required to eliminate the displacement incompatibility caused by the membrane solutions alone.

In creep, this is precluded because of the nonlinear material behaviour, but a similar approach can be followed if it is recognized that at the final steady stress state the strain rates exactly annihilate those due to the displacement incompatibility of the membrane solutions considered in isolation.

The method which can be applied to any component for which an elastic solution is available, is illustrated by consideration of a simple beam with pure bending under creep conditions and is then developed for several different pressure vessel components.

The time variation of stresses from their initial values to their final steady states and the rate of accumulation of damage in that time are developed.

INTRODUCTION

Many thin shell structures have been analyzed for elastic conditions by making use of the membrane solutions for the component parts to find the displacement incompatibility caused by the membrane solutions alone.

Edge moments and forces which form a self-equilibrating system can then be applied to each component to counteract the displacement incompatibility of the membrane solutions and superposition of the results gives the total elastic solution.

The present work applies the same sort of reasoning to the case where the structure operates in the creep regime and the elastic solution is known.

Under creep conditions the displacement incompatibility of the membrane solution alone increases with time, and simultaneously creep strains which generally reduce the elastic discontinuity strains occur so that there will be a redistribution of elastic strains and therefore elastic stresses.

The discontinuity strains are defined by the elastic solution together with the creep strains due to the membrane solutions alone but the amounts of creep strain which occur

depend on the current state of stress and are therefore more difficult to quantify.

However, eventually the system reaches a steady state when no further redistribution of stress occurs at which point the creep strain rates must exactly annihilate the discontinuity strain rates due to the membrane solutions acting alone.

Therefore, if the discontinuity strains ϵ_d are considered proportional to the elastic membrane incompatibility D , and under creep conditions the membrane strain rates are $\dot{\epsilon}_{memc}$, and the rate of change of the membrane incompatibility is $d/dt(D)$, the steady state creep strain rates $\dot{\epsilon}_c$ are given by,

$$\dot{\epsilon}_c = \frac{d/dt(D)}{D} \epsilon_d + \dot{\epsilon}_{memc}$$

Then the steady state stresses and other parameters can be derived.

BEAM UNDER UNIFORM BENDING

Let, $2h$ = beam depth M = applied moment

σ = stress x = distance from neutral axis

The elastic stress distribution is

$$\sigma = 1.5 M/h^2 (x/h)$$

and, if the equation governing creep is described by,

$$\dot{\epsilon} = f(t) \sigma^n,$$

it can be shown that the creep stress distribution can be described by:

$$\sigma_c = \frac{(2n+1)}{2n} M/h^2 (x/h)^{1/n} \quad (1)$$

For the case of uniform bending the beam may be considered to be two separate halves where the 'membrane' stress σ_{mem} in each half is the uniform stress which will provide equilibrium with the applied load.

$$\sigma_{mem} = M/h^2$$

The strain in the upper half is the negative of that in the bottom half so that the difference in the strains is:

$$\Delta \epsilon_{mem} = 2M/(Eh^2)$$

The discontinuity stresses are proportional to this difference.

$$\begin{aligned} \sigma_{disc} &= \sigma_{elastic} - \sigma_{mem} \\ &= (1.5(M/h^2)(x/h) - M/h^2) \\ &= (M/h^2)(1.5(x/h) - 1) \end{aligned}$$

Under creep conditions the membrane stresses stay the same but in the top half the creep strain rate under the membrane stress is given by:

$$\dot{\epsilon}_{mem} = f(t) (M/h^2)^n$$

and the difference in the strain rates in the top and bottom halves is:

$$\Delta \dot{\epsilon}_{mem} = 2f(t) (M/h^2)^n$$

Therefore discontinuity stresses proportional to this difference will be induced at the same rate and simultaneously, creep strains defined by the current values of the stresses and generally reducing the strains due to the discontinuity stresses, will be induced.

Stress redistribution will proceed until steady conditions are reached when the strains due to the elastic discontinuity stresses, induced at a rate defined by the constant membrane stress solution, plus the creep strains due to the membrane stresses, are exactly annihilated by the creep strains caused by the steady state stress distribution.

The steady state creep strain rate is therefore:

$$\begin{aligned} \dot{\epsilon}_{creep} &= \dot{\epsilon}_{mem} + \frac{\Delta \dot{\epsilon}_{mem}}{\Delta \epsilon_{mem}} (1/E(\sigma_{disc})) \\ &= f(t) (M/h^2)^n + \frac{2f(t) (M/h^2)^n M/Eh^2 (1.5x/h - 1)}{2M/(Eh^2)} \end{aligned}$$

$$= f(t) (M/h^2)^n (1 + 1.5x/h - 1)$$

$$= f(t) (M/h^2)^n (1.5x/h)$$

Therefore:

$$\sigma_{creep} = M/h^2 (1.5x/h)^{1/n}$$

and comparing this with equation (1) gives:

$$\frac{\sigma_{creep}}{\sigma_c} = \frac{2n(1.5)^{1/n}}{(2n+1)} \quad (2)$$

The ratio is within two percent of unity for all values of n so that the two methods can be considered to produce the same results. **BUTTING CYLINDERS OF UNEQUAL THICKNESS WITH INTERNAL PRESSURE**

Elastic Solution

The elastic membrane solution for each of the two cylinders of radius R under internal pressure P is:

$$\begin{aligned} \sigma_{HOOP} &= PR/T & \sigma_{hoop} &= PR/t \\ \sigma_{LONG} &= PR/2T & \sigma_{long} &= PR/2t \end{aligned}$$

where T is the thickness of the thicker cylinder and t is that of the thinner cylinder. The difference in the membrane radial displacements of the cylinders is:

$$\Delta c_{yl} = (PR/E)(1-\nu/2)(R/t - R/T) \quad (3)$$

Discontinuity moments and forces induced to overcome this difference and their variation with non dimensional distance from the junction Bx are readily calculated by thin shell theory (2).

The stress concentration factor k is obtained by dividing the stress by PR/t , the hoop membrane stress in the thinner cylinder so that the hoop and longitudinal stress concentration factors in the thinner cylinder can be written as:

$$kh = 1 + khd \quad kl = 0.5 + kld \quad (4)$$

where khd and kld are the hoop and longitudinal components due to the discontinuity moments and forces and are proportional to Δc_{yl} .

The discontinuity strains can be found from:

$$\epsilon_{hd} = (PR)/(tE) ((1-\nu k_l/k_h)kh - (1-\nu/2)) \quad (5)$$

$$\epsilon_{ld} = (PR)/(tE) ((1-\nu k_h/k_l)kh(kl/kh) - (1/2-\nu))$$

Creep Solution

Under creep conditions, the membrane stresses are the same as in the elastic case but the membrane displacements are different. etc.

The constitutive equation which is used to describe creep is, in general:

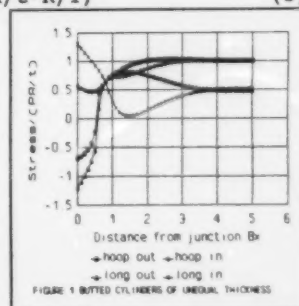


FIGURE 1 BUTTED CYLINDERS OF UNEQUAL THICKNESS

$$\dot{\epsilon} = f(t) \sigma^n \quad (6)$$

such that:

$$\begin{aligned} \dot{\epsilon}_1 &= (\sigma_1 - (\sigma_2 + \sigma_3)/2) \sigma_{\text{bar}}^{(n-1)} f(t) \\ \sigma_{\text{bar}} &= ((\sigma_1)^2 - (\sigma_1)(\sigma_2) + (\sigma_2)^2)^{0.5} \end{aligned} \quad (7)$$

The rate of change of the difference in radial displacement of the two cylinders under membrane conditions is:

$$d/dt(D_{\text{cyl}}) = (3/4)^{(n+1)/2} (PR)^n ((1/t)^n - (1/T)^n) f(t) R \quad (8)$$

The steady state creep strain rates are:

$$\begin{aligned} \dot{\epsilon}_{\text{hc}} &= \dot{\epsilon}_{\text{hd}} d/dt(D_{\text{cyl}}) + \dot{\epsilon}_{\text{hm}} & \dot{\epsilon}_{\text{lc}} &= \dot{\epsilon}_{\text{ld}} d/dt(D_{\text{cyl}}) + \dot{\epsilon}_{\text{lm}} \\ D_{\text{cyl}} & & D_{\text{cyl}} & \end{aligned} \quad (9)$$

Note that the membrane creep rates in the thinner cylinder are:

$$\begin{aligned} \dot{\epsilon}_{\text{hm}} &= (3/4)^{(n+1)/2} (PR/t)^n f(t) R \\ \dot{\epsilon}_{\text{lm}} &= 0 \end{aligned} \quad (10)$$

Combining equations (3), (5), (8) and (10) in equation (9) gives:

$$\dot{\epsilon}_{\text{hc}}/\dot{\epsilon}_{\text{lc}} = \frac{(k_{\text{hd}} - v k_{\text{ld}})}{(k_{\text{ld}} - v k_{\text{hd}})} + \frac{(1-v/2)(1-t/T)}{(k_{\text{ld}} - v k_{\text{hd}})(1-(t/T)^n)} \quad (11)$$

From the general creep relationships:

$$\sigma_{\text{lc}}/\sigma_{\text{hc}} = \frac{(2 + \dot{\epsilon}_{\text{hc}}/\dot{\epsilon}_{\text{lc}})}{(2 + \dot{\epsilon}_{\text{lc}}/\dot{\epsilon}_{\text{lc}} + 1)} \quad (12)$$

Finally:

$$(k_{\text{hc}})^n = (3/4)^{(n+1)/2} \left\{ \frac{1 + (k_{\text{hd}} - v k_{\text{ld}})(1-(t/T)^n)}{(1-t/T)(1-v/2)} \right\} \quad (13)$$

$$\frac{1 - (1/2)\sigma_{\text{lc}}/\sigma_{\text{hc}}}{(1 - \sigma_{\text{lc}}/\sigma_{\text{hc}} + (\sigma_{\text{lc}}/\sigma_{\text{hc}})^2)^{(n-1)/2}} \quad (13)$$

The creep stresses at the inside and outside surfaces at positions along the thinner cylinder for the case $T/t \gg 1$ and $n = 3$ are shown in FIGURE 1

Weld of Low Creep Strength

By allowing the weld to have a similar constitutive relationship to that of the cylinder material, a relatively simple solution can be obtained for the case where the weld material is significantly less creep resistant than the parent material. The weld material is assumed to follow:

strain rate = $g(t)$ (stress)^m
and by allowing $g(t)/f(t)$ and m/n to vary the effect of different weld strengths can be investigated as shown for an example in FIGURE 2.

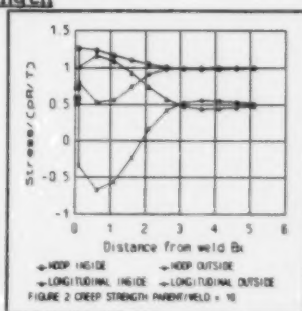


FIGURE 2 CREEP STRESS PARENT/WELD = 10

CYLINDER SPHERE JUNCTION WITH INTERNAL PRESSURE

Elastic Solution

The membrane solution for the sphere is:

$$\sigma_{\text{hoop}} = (PR/2T) \quad (14)$$

$$\sigma_{\text{long}} = (PR/2T)$$

and the radial displacement at the intersection is,

$$\begin{aligned} \text{disp sphere} &= r/E ((PR/2T) - v (PR/2T)) \\ &= r (1-v) (PR/2TE) \end{aligned} \quad (15)$$

For the cylinder, the membrane stresses are in equilibrium with the internal pressure and also with the external pressure on the outside surface of the cylinder at the intersection over a length equal to the shell thickness which is necessary to provide equilibrium with the shell membrane stresses. The external pressure is P_B where,

$$B = (R/2T) (1 - (r/R)^2)^{(1/2)} \quad (16)$$

Then the displacement of the cylinder can be written as,

$$\text{disp cyl} = r((1+B) - v/2) (Pr/tE) \quad (17)$$

The discontinuity stresses induced at the junction are proportional to the difference in the membrane displacements. That is, from equations (15) and (16) and denoting the difference in the membrane displacements by D_{sh}

$$\begin{aligned} D_{\text{sh}} &= r(1-v) (PR/2TE) - r((1+B) - v/2) (Pr/tE) \\ &= r(PR/2TE) ((1-v) - 2((1+B) - v/2) (r/R) (T/t)) \end{aligned} \quad (18)$$

Denoting the stresses induced at the position of maximum stress by:

$$\sigma_{\text{hoop}} = k_h (PR/2T); \quad \sigma_{\text{long}} = k_l (PR/2T) \quad (19)$$

the discontinuity stresses σ_{hd} and σ_{ld} are:

$$\sigma_{\text{hd}} = (PR/2T) (k_h - 1); \quad \sigma_{\text{ld}} = (PR/2T) (k_l - 1) \quad (20)$$

and the elastic discontinuity strains are:

$$\begin{aligned} \epsilon_{\text{hd}} &= (PR/2TE) ((1 - v k_l/k_h) k_h - (1-v)) \\ \epsilon_{\text{ld}} &= (PR/2TE) ((1 - v k_h/k_l) k_h (k_l/k_h) - (1-v)) \end{aligned} \quad (21)$$

Creep Solution

The membrane displacements vary with time such that for the sphere:

$$\begin{aligned} \text{rate of disp} &= (PR/2T)^n (1-1/2) (1+1-1)^{(n-1)/2} r f(t) \\ &= (r/2) f(t) (PR/2T)^n \end{aligned} \quad (22)$$

for the cylinder:

$$\text{rate of disp} = (Pr/t)^n (3/4 + B) (3/4 + (3/2)B + B^2)^{(n-1)/2} r f(t) \quad (23)$$

Defining the elastic discontinuity strains as ϵ_{hd} , ϵ_{ld} , the membrane creep rates as $\dot{\epsilon}_{\text{hm}}$, $\dot{\epsilon}_{\text{lm}}$ and the final creep strain rates as $\dot{\epsilon}_{\text{hc}}$, $\dot{\epsilon}_{\text{lc}}$,

$$\begin{aligned}\dot{\epsilon}_{hc} &= \frac{\dot{\epsilon}_{hd}}{D_{sh}} \frac{d}{dt}(D_{sh}) + \dot{\epsilon}_{hm} \\ \dot{\epsilon}_{lc} &= \frac{\dot{\epsilon}_{ld}}{D_{sh}} \frac{d}{dt}(D_{sh}) + \dot{\epsilon}_{lm}\end{aligned}\quad (24)$$

For the spherical shell, combining equations 3.5, 3.11 and 3.12 gives:

$$\frac{d}{dt}(D_{sh}) = \frac{f(t)(PR/2T)^n}{D_{sh}(PR/2TE)^2} f(B) \quad (25)$$

where,

$$f(B) = \frac{(1-2(3/4+B)(3/4+(3/2)B+B^2))^{((n-1)/2)}((r/R)(2T/t))^n}{((1-v)-((1+B)-v/2)((r/R)(2T/t)))} \quad (26)$$

Noting that for the shell,

$$\dot{\epsilon}_{hm} = \frac{f(t)(PR/2T)^n}{2}$$

$$\dot{\epsilon}_{lm} = \frac{f(t)(PR/2T)^n}{2} \quad \text{and using equations (21),}$$

$$\dot{\epsilon}_{hc} = (1/2)f(t)(PR/2T)^n(1+(khd-vkld)f(B)) \quad (27)$$

$$\dot{\epsilon}_{lc} = (1/2)f(t)(PR/2T)^n(1+(kld-vkhd)f(B))$$

In general terms from the constitutive relations (6) and (7),

$$\begin{aligned}\dot{\epsilon}_{hc} &= f(t)(\sigma_{hc} - (1/2)\sigma_{lc})(\sigma_{hc}^2 + \sigma_{lc}^2 - \sigma_{hc}\sigma_{lc})^{((n-1)/2)} \\ \dot{\epsilon}_{lc} &= f(t)(\sigma_{lc} - (1/2)\sigma_{hc})(\sigma_{hc}^2 + \sigma_{lc}^2 - \sigma_{hc}\sigma_{lc})^{((n-1)/2)}\end{aligned}\quad (28)$$

Note from equations (28) that,

$$\sigma_{lc}/\sigma_{hc} = \frac{(2+\dot{\epsilon}_{hc}/\dot{\epsilon}_{lc})}{(2\dot{\epsilon}_{hc}/\dot{\epsilon}_{lc}+1)} \quad (29)$$

Manipulating equations (28) gives:

$$\dot{\epsilon}_{hc} + (1/2)\dot{\epsilon}_{lc} = f(t)(3/4)(\sigma_{hc})(\sigma_{hc}^2 + \sigma_{lc}^2 - \sigma_{hc}\sigma_{lc})^{((n-1)/2)}$$

That is,

$$(khc)^n = \frac{(\dot{\epsilon}_{hc} + (1/2)\dot{\epsilon}_{lc})}{(3/4)f(t)(PR/T)^n(1+(\sigma_{lc}/\sigma_{hc})^2 - (\sigma_{lc}/\sigma_{hc}))^{((n-1)/2)}} \quad (30)$$

Using the results presented in (4) for the maximum elastic stress in the shell, and assuming $kl/kh = 0$ for elastic conditions, the above method allows the steady state creep stresses to be found.

The results are shown in FIGURE 3 for one value of the parameter t/T . It can be seen that for some cases the value of khc is less than the elastic value of kh but for other cases it is greater suggesting that designing for creep conditions based only on elastic considerations carries some risk.

It must be noted that the ratio kl/kh for the elastic solution has been assumed to be zero in the examples in FIGURE 3 so that although the trends shown are correct, the absolute values will depend on the complete elastic solution and may be different than those shown.

CYLINDER-CYLINDER INTERSECTION WITH INTERNAL PRESSURE

The cylinder-cylinder geometry can be treated in a similar way if the elastic

solution is known although the difference in the membrane displacements of the two components varies around the circumference.

It has been reported (5) that the position of the maximum stress in the cylinder-cylinder intersection lies

in the shell on the longitudinal section and therefore the elastic discontinuity stresses are assumed to be proportional to the difference in the membrane displacements at the longitudinal section.

A similar set of equations as were developed for the cylinder-sphere junction can then be developed for the cylinder-cylinder intersection.

$$\frac{d}{dt}(D_{cyl}) = -\frac{(PR/T)^n f(t) f(B)}{(PR/T)} \quad \text{where,} \quad (31)$$

$$f(B) = \frac{(3/4+B)(3/4+(3/2)B+B^2)^{((n-1)/2)}((r/R)(T/t))^n}{((1/2-v)-(r/R)(T/t)(1-v/2+B))} \quad (32)$$

$$\dot{\epsilon}_{hc}/\dot{\epsilon}_{hmc} = \frac{1-f(B)(kh-vkl)-1+v/2}{(3/4)((n+1)/2)} \quad (33)$$

$$\dot{\epsilon}_{lc}/\dot{\epsilon}_{hmc} = \frac{0-f(B)(kl-vkh)-1/2+v}{(3/4)((n+1)/2)}$$

$$\sigma_{lc}/\sigma_{hc} = \frac{(2+\dot{\epsilon}_{hc}/\dot{\epsilon}_{lc})}{(2\dot{\epsilon}_{hc}/\dot{\epsilon}_{lc}+1)} \quad (34)$$

$$(khc)^n = \frac{(\dot{\epsilon}_{hc} + (1/2)\dot{\epsilon}_{lc})}{(3/4)f(t)(PR/T)^n(1+(\sigma_{lc}/\sigma_{hc})^2 - (\sigma_{lc}/\sigma_{hc}))^{((n-1)/2)}} \quad (35)$$

The group $(r/R)(T/t)$ appears in the above equations and this can be rearranged as follows:

$$\text{Let } \alpha = (r/t_{min})/(R/T_{min})$$

where t_{min}, T_{min} are the minimum thicknesses of the nozzle and shell respectively.

For $F_s = T/T_{min}$ = shell thickening factor

and $F_n = t/t_{min}$ = nozzle thickening factor,

$$(r/R)(T/t) = \alpha F_s/F_n$$

FIGURE 4 shows the variation of khc with R/T for the parameter $\alpha F_s/F_n$ using the relationship in (5) between kh and $(r/t)^2(T/R)$ using the creep index $n = 7$.

The difference between the results decreases with incremental increases in the creep index so that the difference between $n' = 7$ and $n' = 100$ is insignificant.

$n' = 100$ corresponds to the limit case where the limit pressure P_l as a fraction of the membrane yield pressure P_{mem} is given by:

$$\frac{P_l}{P_m} = \frac{1}{khc}$$

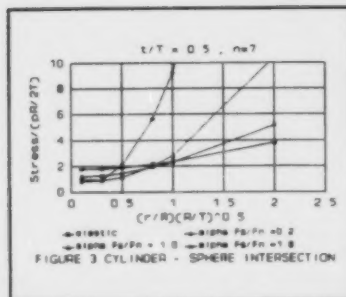
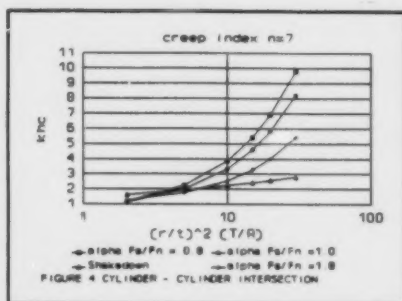


FIGURE 3 CYLINDER - SPHERE INTERSECTION

It has been suggested (6) that the limit pressure may be used as an estimate of the creep rupture life of a component if the uniaxial stress to produce rupture in a set time is substituted for the yield stress of the material. The present results indicate agreement with this if the creep index is at least 'n'=7.



The concept of shakedown has been used in the design of pressure vessels operating at elevated temperature (7) and in this case the shakedown factor was taken to be 90 percent of the theoretical value derived from the results of (5) treated in the manner described in (8).

This method essentially assumes that the value of k_{hc} will be related to the initial elastic stress concentration factor k_h and the elastic shakedown factor k_s suitably modified by the 90 percent factor in the following way:

$$k_{hc} = \frac{k_h}{(0.9k_s)}$$

Using the relationships of (5) and (8), the variation of k_{hc} assumed in the "shakedown" analysis can be obtained and compared to the values predicted by the present method. FIGURE 4 shows that for values of $(r/t)^2(T/R)$ below about 4 the "shakedown" approach is more conservative than the present method but for higher values the reverse is true and the "shakedown" method would yield optimistic results compared to the present method.

REDISTRIBUTION TIME

The final steady creep stresses at the position of maximum elastic stress have been found in the above and the variation of these stresses with time can also be estimated.

Noting that the total strain rate at a point is equal to the strain rate corresponding to the steady creep solution, the rate of stress redistribution can be found.

That is:

$$\begin{aligned} \dot{\epsilon}_h + \dot{\epsilon}_{hr} &= \dot{\epsilon}_{hc} ; \dot{\epsilon}_l + \dot{\epsilon}_{lr} = \dot{\epsilon}_{lc} \text{ where,} \\ \dot{\epsilon}_h &= \text{hoop creep strain rate due to } \sigma_h, \sigma_l \\ \dot{\epsilon}_l &= \text{long creep strain rate due to } \sigma_h, \sigma_l \\ \dot{\epsilon}_{hr} &= \text{elastic hoop strain rate} \\ \dot{\epsilon}_{lr} &= \text{elastic long strain rate} \\ \dot{\epsilon}_{hc} &= \text{steady hoop creep strain rate} \\ \dot{\epsilon}_{lc} &= \text{steady long creep strain rate} \end{aligned}$$

By rearranging, the rate of change of elastic stress due to the elastic strain rates can be obtained,

$$\begin{aligned} d\sigma_h/dt &= E/(1-\nu^2) (\dot{\epsilon}_{hc} + \nu\dot{\epsilon}_{lc} - \dot{\epsilon}_h - \nu\dot{\epsilon}_l) \\ d\sigma_l/dt &= E/(1-\nu^2) (\dot{\epsilon}_{lc} + \nu\dot{\epsilon}_{hc} - \dot{\epsilon}_l - \nu\dot{\epsilon}_h) \end{aligned} \quad (36)$$

By manipulating equations (36) it is possible to show that,

$$\frac{\dot{\epsilon}_{hc} dt}{(PR/TE)} = \frac{(1 - \nu(d\sigma_l/d\sigma_h)) dk_h}{(1 - (\dot{\epsilon}_h/\dot{\epsilon}_{hc}))} \quad (37)$$

Recall that,

$$\dot{\epsilon}_{hc} = (3/4)^{((n+1)/2)} (PR/T)^n f(t)$$

So that,

$$\frac{\dot{\epsilon}_{hc} dt}{(PR/TE)} = \frac{(3/4)^{((n+1)/2)} (PR/T)^n f(t) dt}{(PR/TE)}$$

and so by defining a new time Σ by,

$$d\Sigma = \frac{(3/4)^{((n+1)/2)} (PR/T)^n f(t) dt}{(PR/TE)}$$

So long as the properties of the real material being considered are known it is possible to plot the variation of k_h and k_l with real time.

$$\frac{\dot{\epsilon}_{hc} dt}{(PR/TE)} = d\Sigma \quad (38)$$

and from equation (37),

$$d\Sigma = \frac{(1 - \nu(d\sigma_l/d\sigma_h)) dk_h}{(1 - (\dot{\epsilon}_h/\dot{\epsilon}_{hc}))} \quad (39)$$

so that by choosing small values for dk_h and repeatedly solving (39), the variation of k_h and k_l with Σ can be found.

In FIGURE 5 for the cylinder-sphere intersection and FIGURE 6 for the cylinder-cylinder intersection, the material is taken to be

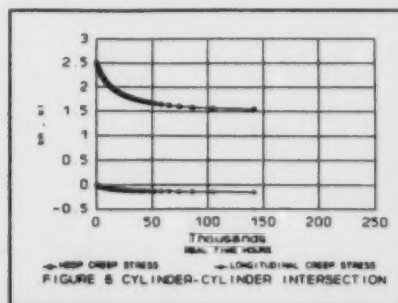
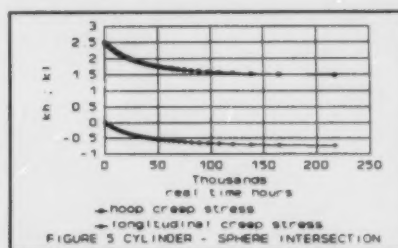
2 1/4% Cr at a temperature of 1000F and the data for this material was taken from (9).

The example shown in FIGURE 5 shows that the redistribution time is about 100000 hours.

For this case the shell membrane hoop

stress was 4400 psi and if this were to be increased to 6300 psi, the redistribution time would reduce to about 15000 hours.

FIGURE 6 for the cylinder-cylinder



cylinder-cylinder intersection with parameters similar to the cylinder-sphere intersection of FIGURE 5 shows that when the shell membrane hoop stress is 4400 psi, the redistribution time is about 60000 hours.

When the stress is increased to 6300 psi, the redistribution time decreases to about 10000 hours.

Whether or not the redistribution time represents a significant portion of the whole life of the component seems to be very case specific.

RUPTURE TIME

It is common to use a linear cumulative damage law to determine the life fraction used during different times at different stress levels in the creep regime.

This procedure is followed here to estimate the accumulated damage during the redistribution time and then in further time at steady creep conditions.

In order to do this, it is necessary to establish a relationship between stress and rupture time. In this, the equivalent stress is taken to the appropriate one and for the material considered a suitable relationship between stress and rupture time was derived from published data (9).

FIGURE 7 shows results for the same 2 1/4 % Cr material and the same geometric parameters as used previously for FIGURE 5

When the shell membrane hoop stress is 4400 psi, the total damage accumulated at the maximum stress position in 200000 hours of service would be 80 percent of the theoretical maximum. In the membrane region of the shell remote from discontinuities the life fraction used in the same time would be about 10 percent.

If the membrane stress were to be increased to 6300 psi, the total damage sustained at the position of maximum stress would be 100 percent at about 25000 hours of service at which time the damage sustained in the membrane region would be about 4 percent.

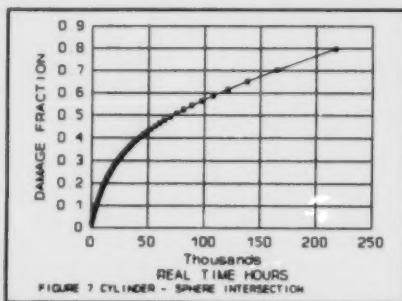


FIGURE 7 CYLINDER - SPHERE INTERSECTION

FIGURE 8 shows the results for the cylinder-cylinder intersection previously treated in FIGURE 6.

When the shell membrane hoop stress is 4400 psi, the total accumulated damage at the position of maximum stress in a service life of 140000 hours would be about 28 percent and for the membrane region in the same time the damage would be about 7 percent.

If the membrane hoop stress were to be increased to 6300 psi, the damage at the maximum stress position would reach 100 percent at about 140000 hours and in the same time the damage in the membrane region would

be about 21 percent.

The conclusion which can be reached is that the damage accumulated during the stress redistribution time and then later at the steady stress condition may significantly reduce the expected life of the component in some cases and have a minor impact in other cases. Each case should be analyzed on an individual basis.

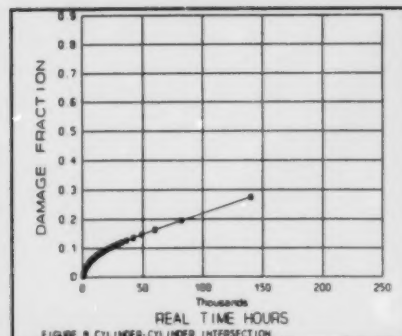


FIGURE 8 CYLINDER-CYLINDER INTERSECTION

CONCLUSIONS

It has been shown that if the elastic solution and the material properties are known for a pressure vessel component, a creep solution can be derived.

The accuracy of the creep solution depends on the accuracy of the elastic solution and the accuracy of the material properties rules used.

Designs based on elastic considerations such as shakedown may not provide the optimum creep life since sometimes the predicted stresses are optimistic.

As the stresses vary from their initial values to their steady state values, damage accumulates at a rate greater than the eventual steady state damage rate. This damage may represent a significant fraction of the total life of the component.

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RECENT DEVELOPMENTS IN THE EVALUATION OF GASKET PERFORMANCE AT ELEVATED TEMPERATURE

by

LUC MARCHAND
ANDRÉ BAZERGUI
MICHEL DERENNE

Section of Applied Mechanics, Department of Mechanical Engineering
École Polytechnique de Montréal
P.O. Box 6079, Station "A", Montréal, Qué., Canada H3C 3A7

ABSTRACT

This paper discusses a recently developed approach for evaluating sheet gasket performance at elevated temperature that is based on gasket weight loss during thermal exposure. Using simple testing devices, strong correlations are shown to exist between weight loss and the change of key gasket properties during aging, namely: tightness, thickness change and tensile strength. These correlations permit a better understanding of the fundamental mechanisms involved in the degradation process of elastomer bound sheet gasket materials and have led to the development of an aging parameter (A_p), based on weight loss, that successfully combines the time and temperature effect.

Through the use of gasket weight loss and A_p , the influence of some important parameters on the degradation rate of the gasket is investigated. These include: aging atmosphere, gasket width, initial compression stress, and the internal gas media. Finally, a method of characterizing elevated temperature gasket performance with a temperature index is proposed.

NOMENCLATURE

A_p = Aging parameter based on gasket weight loss, $h^{0.6}/^{\circ}K$
 $A_{p\max}$ = Value of A_p at maximum tolerable weight loss
 $L_{rm\ eq}$ = Equivalent Helium mass leak rate, mg/s
 t = exposure time, h
 T = exposure absolute temperature, $^{\circ}K$
 T_0 and T_0^* = temperature constants, $^{\circ}K$
 $T_{op.\max}$ = Predicted maximum operating temperature
 w = fraction of gasket weight loss
 $w_{\max.\ tol.}$ = maximum tolerable weight loss

INTRODUCTION

Piping systems and other pressurized equipments are usually fitted with gasketed bolted flanged connections. In order to prevent costly and dangerous leakage, these connections and especially those operating at elevated temperature most rely on an efficient

gasket whose performance must be stable over long periods of time. The evaluation of gasket behavior under long term exposure at elevated temperature is thus of prime importance for manufacturers and users of gasket products.

One of the most common types of sheet gasket is compressed fiber reinforced elastomer sheets. These sheets usually consist of elastomer binder, reinforcing fibers and inorganic fillers. Through years of testing it has been observed that elastomer based sheet gaskets when subjected to elevated temperature exposure over time undergo changes in their physical properties and their final appearance. It was also noticed that these changes are always coupled with weight loss of the gasket material.

Since weight loss due to thermal exposure is somehow directly related to chemical reactions involved in the degradation process, a correlation is expected to exist between changes in gasket key physical properties (e.g. sealability, creep/relaxation, tensile strength) and weight loss of the material. By using a time-temperature correlation based on gasket weight loss it is also possible to obtain a simple predictive parameter that has potential for becoming a powerful tool for the evaluation of gasket performance at elevated temperature. The study of the influence of some important parameters on the degradation rate of the gasket can also be achieved through the use of gasket weight loss and this predictive parameter. This paper summarizes the outcome of work recently performed in this field.

TEST METHODS

Four different tests methods have been used throughout this study. Most of the weight loss correlations were obtained using two simple mechanical testing devices, the ATRS device and the more recent ARLA device, which are described below. However, in order to measure the influence of the internal gas media on the degradation rate of the gasket, comparisons were also made with

results obtained from two types of elevated temperature tightness tests (HOTT and AHOT).

Mechanical Devices

The ATRS (Aged Tensile Relaxation Screen test, [1]) fixture is a spring loaded fixture used to age rectangular gasket specimens under load and creep/relaxation conditions representative of real flange operating conditions (Fig. 1). Thermal exposure is performed in a still air oven. The gasket properties obtained after aging are: residual tensile strength, load relaxation, thickness change, and gasket weight loss.

The ARLA (Aged Relaxation Leakage Adhesion screen test device, [2]) fixture retains the mechanical characteristics of the ATRS fixture but uses one circular gasket sample (Fig. 1). The gasket properties obtained after aging are: leak rate, weight loss and load relaxation. Gasket leak rate measurements are performed, before and after aging, by applying an internal Helium pressure of 5.5 MPa. Thermal aging can be achieved by placing the ARLA fixtures in still air ovens or in special sealed chambers adapted to ovens. The degradation process in an inert atmosphere of Nitrogen was investigated using these chambers.

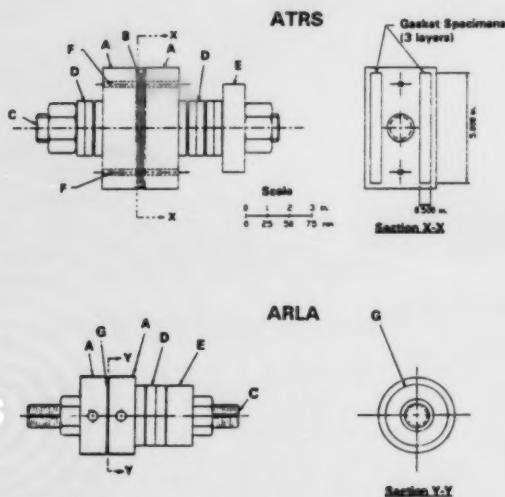


Fig. 1 ATRS and ARLA fixtures: (A) compression platens, (B) stacked rectangular specimens, (C) stud bolt, (D) springs, (E) load collar, (F) pins for thickness measurements, (G) circular specimen.

Elevated Temperature Tightness Tests

The HOTT (Hot Operational Tightness Test, [3]) test involves initial compression at room temperature (simulation of Bolt-up), aging (3 days or more) under creep-relaxation conditions, and stress excursions and thermal disturbance cycles. During these tests, 4 in. (102 mm) NPS gaskets are subjected to thermal aging under a sustained 5.5 MPa internal gas pressure.

The AHOT (Aged Hot Operational Tightness test, [4]) is an alternative to the HOTT procedure for situations requiring long aging periods. It differs from the HOTT procedure only during the aging period whereby the precompressed gasket/platen assembly is aged in a kiln instead of the HOTT fixture. Because of this difference, the AHOT aging period is performed under a reduced internal gas pressure of 69 kPa.

SHEET GASKET MATERIALS

Many of the test results presented in this paper were obtained with two types of sheet gasket materials, a compressed Asbestos (A8) and a compressed Aramid sheet (N5), both made by the same manufacturer. Testing was also performed on two other compressed Aramid sheets (N1, N6) and on one compressed Fiber-Glass sheet (N7). All these materials had the same NBR binder and the same initial thickness of 1.6 mm.

WEIGHT LOSS CORRELATIONS

In Fig. 2, the change in three key gasket properties of an NBR-Aramid sheet (N5) is plotted as a function of the gasket weight loss. These properties include thickness change (creep), leak rate, and post aging tensile strength. Strong correlations are observed between property changes and gasket weight loss particularly in the case of leak rate and creep.

By analyzing the change in gasket properties as a function of weight loss during thermal aging in air and inert Nitrogen atmospheres, it is possible to identify three different phases in the thermal degradation process of the gasket material:

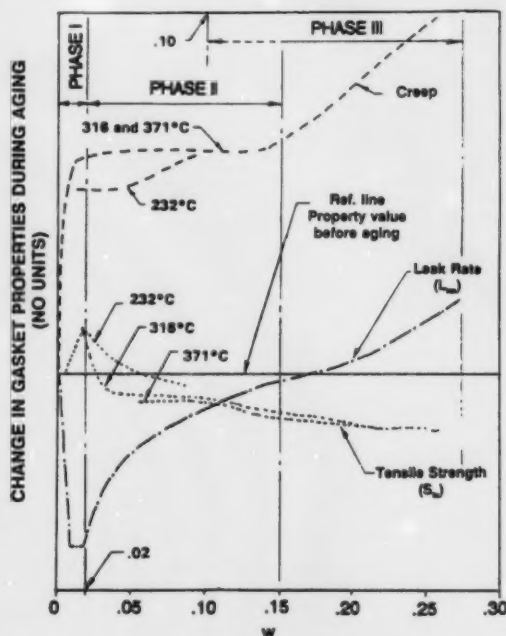


Fig. 2 Change in gasket properties as a function of gasket weight loss during air aging. NBR-Aramid (N5) sheet material.

- Phase I, between 0 and 2% of weight loss, is characterized by the transformation of the NBR binder. When heat is initially applied, the binder first softens causing the gasket to creep substantially. As a result, gasket density reaches its maximum thus reducing material porosity as shown by the leak rate curve which is at its minimum.
- Phase II, between 2% and 15% of weight loss. Here, pyrolysis and oxidation of the NBR binder dominate and a gradual deterioration of the gasket properties occurs.
- Phase III, starting at about 10% or 15% of weight loss. This is characterized by the degradation of the aramid fibers and a rapid deterioration of the gasket properties as a function of weight loss. This third phase of degradation does not exist when reinforcing fibers can sustain exposure temperatures as it the case with the NBR-asbestos (A8) sheet.

One of the most interesting correlation obtained so far is the one between the gasket weight loss and its leak rate after aging. In Fig. 3, the room temperature leak rate is plotted against weight loss of NBR-asbestos (A8) and NBR-Aramid (N5) sheet gaskets. For both materials, thermal exposures were performed under several time and temperature combinations as well as in different aging atmospheres including: oxidising (air), non-oxidising (nitrogen), and a combination of both (nitrogen aging followed by air aging). The results presented in Fig. 3 indicate that, for both materials, the leak rate is a direct function of the weight loss. Also, both materials exhibit an almost identical leak rate curve until weight loss reaches 16%. This is the maximum weight loss for the asbestos sheet gasket. Past this point, the aramid gasket leak rate continues to increase in proportion of its weight loss.

From these observations, we can draw two partial conclusions. First, the change in leak rate is a function only of gasket weight loss and is not influenced by the aging conditions (i.e. gas media, temperature). Second, the evolution of leak rate as a function of degradation appears to be identical for sheet gasket materials of similar types.

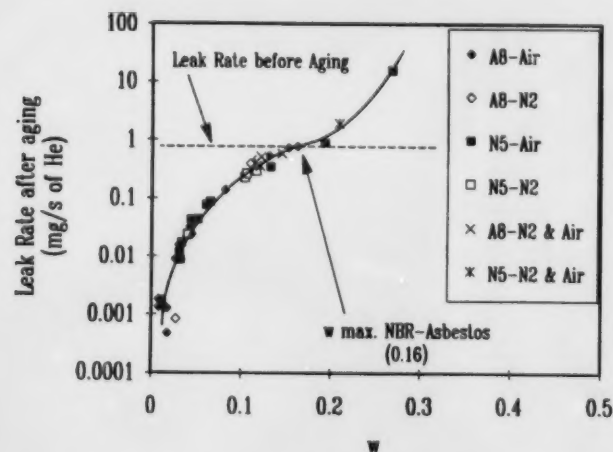


Fig. 3 Leak rate after aging as a function of gasket weight loss. NBR-Asbestos (A8) and NBR-Aramid (N5) sheet materials.

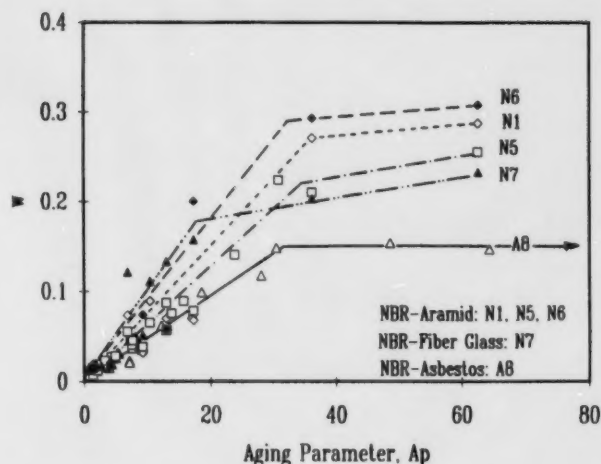


Fig. 4 Weight loss as a function of aging parameter (A_p).

THE NEW AGING PARAMETER, A_p

In order to develop an aging parameter based on gasket weight loss [6], isothermal and isochronous ATRS tests were conducted on two sheet materials (A8 and N5). An aging parameter, A_p , was derived from the empirical equations of gasket weight loss as a function of time and temperature. A_p has two forms, taking into account the considerable increase in the degradation rate that occurs above 330°C.

$$A_p = 1000 t^{0.6} \left(\frac{1}{T_0} - \frac{1}{T} \right) \quad (185^\circ\text{C} < T < 330^\circ\text{C}) \quad (1)$$

$$A_p = 17000 t^{0.6} \left(\frac{1}{T_0} - \frac{1}{T} \right) \quad (T > 330^\circ\text{C}) \quad (2)$$

The fact that A_p is a function of $t^{0.6}$, (i.e. almost $\sqrt{t}$), indicates that the decomposition process is partly diffusion-controlled [6]. Parameters like gasket width and compression stress will thus influence the degradation rate of gasket materials because they change the length of the diffusion path and the porosity of the material. The influence of these parameters will be discussed in details later in the text.

In Fig. 4, weight loss data obtained with five different types of NBR sheet gaskets are plotted as a function of A_p . An excellent correlation is observed between w and A_p for the five materials, and it can be partly concluded that, within the limits of the presently available data, A_p successfully combines time and temperature to represent the degradation process based on gasket weight loss.

PARAMETERS INFLUENCING THE DEGRADATION RATE

Through the use of gasket weight loss and A_p , the influence of some important parameters on the degradation rate of the gaskets have been investigated.

Effect of Oxidising vs Non-oxidising Atmosphere

Thermal exposure in air and nitrogen atmospheres were performed at several times and temperatures using the ARLA fixture. Weight loss results are plotted as a function of the aging parameter A_p in Fig. 5. Both gasket results indicate that weight loss levels off to a maximum of about 11% when the gasket specimens are aged in Nitrogen.

Weight loss obtained under Nitrogen exposure is believed to be caused mainly by pyrolysis of the rubber binder and possibly other organic components present in the sheet material. Since both types of gasket yielded an identical maximum weight loss of about 11% in Nitrogen, it can be concluded that their NBR binder content is quite similar. For the NBR-Asbestos gasket, the difference between the maximum weight recorded during air exposure (16%) and Nitrogen exposure (11%) can be explained by the oxidation of the carbon residue left after the pyrolysis of the rubber binder is completed. In the case of the NBR-Aramid gasket, the difference between the 30% maximum weight loss for air exposure and the 11% obtained under Nitrogen exposure is much higher. It can only be explained by some additional oxidation reactions of the aramid fibers.

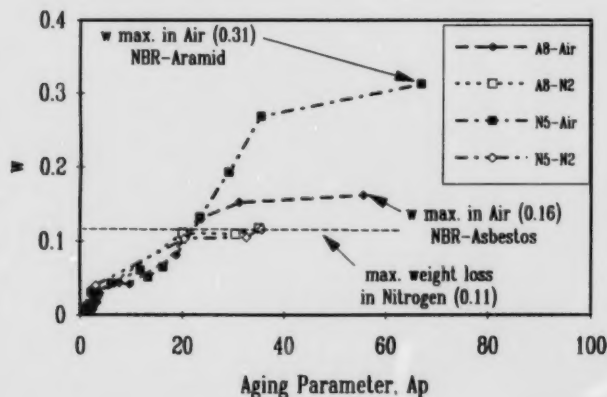


Fig. 5 Effect of aging atmosphere on gasket weight loss. NBR-Asbestos (A8) and NBR-Aramid (N5) sheet materials.

Effect of Gasket Width

The effect of gasket width on the thermal degradation rate of sheet gaskets has been investigated by performing 48-hour aging test with the ARLA fixture at two different temperatures. Results given in Fig. 6 show an increase in degradation rate as the width of the gasket decreases.

Narrowing the gasket reduces the length of the diffusion path so that pyrolysis and oxidation gaseous by-products created during the thermal degradation can escape more rapidly to the atmosphere thus increasing the rate of weight loss.

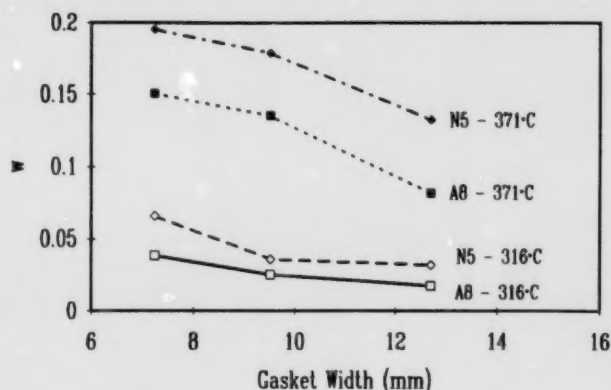


Fig. 6 Effect of width on gasket weight loss during air aging. NBR-Asbestos (A8) and NBR-Aramid (N5) sheet materials.

Effect of the Initial Compressive Stress

To study the effect of the initial compressive stress on the degradation rate, different compressive stresses were initially applied to 9.5 mm and 12.7 mm wide gaskets. Thermal exposures of 48 hours were performed with ARLA fixtures on (A8) and (N5) sheet materials.

Results given in Fig. 7 show that gasket weight loss decreases as the initial compression stress increases. When the compressive stress increases, the internal voids shrink in size, therefore, reducing the porosity of the gasket material. For a gasket undergoing thermal degradation, the ability of pyrolysis and oxidation by-products to escape to the atmosphere depend on the porosity of the gasket and thus on the initial compression stress level of the gasket. As a consequence, a gasket aged under low compressive stress will degrade more rapidly.

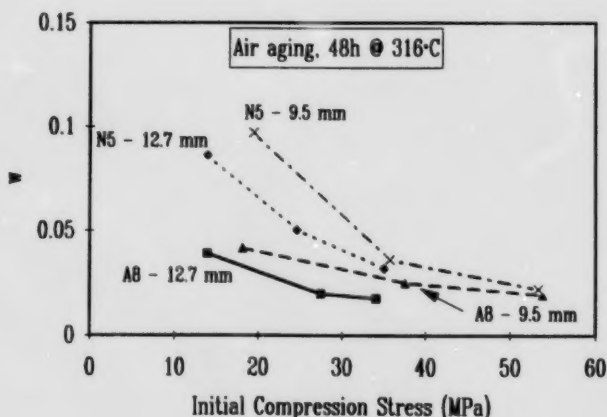


Fig. 7 Effect of initial compression stress on gasket weight loss during air aging. Tests performed on two different gasket widths (9.5 and 12.7 mm). NBR-Asbestos (A8) and NBR-Aramid (N5) sheet materials.

Effect of the Internal Gas Medium

To determine the effect of the internal gas medium upon the degradation rate of a gasket, a comparison was made between ARLA aging tests performed in air atmosphere and previous hot tightness tests (HOTT and AHOT) conducted with air and Helium as the internal pressurized medium. For this study, a NBR-Aramid sheet material (N1) was used. The comparison between ARLA, HOTT and AHOT test results requires some adaptation to account for the different gas media, gasket diameters, exposure times and temperatures. A detailed description of the necessary conversions can be found in [7]. The resulting comparisons are presented in Fig. 8, where the equivalent Helium leak rates ($L_{He eq}$) are plotted as a function of the aging parameter A_p . Because it combines time and temperature effects, A_p is used in this figure to represent the extent of exposure of the gasket.

From Fig. 8, the following observations can be made:

1) The presence of internal air pressure accelerates significantly the degradation process of the gasket material. This can be seen by comparing the leak rate curve of the HOTT test with air (338°C) with the leak rate curve of the ARLA tests. The exact mechanism by which internal air pressure accelerates the degradation process is not yet known but the following explanation seems logical: When leak rate of the HOTT gasket becomes high enough, the continuous flow of air through the gasket might boost the oxidation reactions by supplying fresh oxygen. As leak rate increases, the degradation accelerates until the gasket material is entirely decomposed (point 1, Fig. 8).

2) The use of Helium as the internal medium slows down the degradation rate of the gasket. This can be observed by comparing results of the HOTT test with pressurized Helium (360°C) with those of the ARLA tests ($A_p > 8$). The fluctuations recorded in the HOTT leak rate curve at A_p values of 28 and 32 (points 2 and 3, Fig. 8) where caused by two stress excursion and thermal disturbance cycles performed at the end of the test. As such, they are not an indication of a sudden change in the degradation rate of the gasket and are not considered in the present analysis (the HOTT at 427°C shows similar cycles). The slow-down in the degradation rate caused by the presence of an internal inert gas (Helium) might be explained by the fact that only the outside edge of the HOTT test gasket is in contact with air. This reduces the rate of oxidation reactions as compared to what happens in the ARLA gasket specimen, which has both the inside and the outside edges in contact with air.

3) The similarities between the AHOT (332°C) and HOTT (360°C) leak rates at $A_p = 35$ curves indicates that the pressure level of an inert gas (Helium) has probably little influence on the degradation rate of the gasket. The internal gas pressure maintained during the aging phase of these two tests was 5.5 MPa for the HOTT and 69 kPa for the AHOT.

4) The same level of maximum gasket degradation can be achieved even in the presence of an internal inert gas (Helium) as indicated by the leak rate level of the HOTT test at 427°C. It should be emphasised, however, that in the presence of an internal inert gas the degradation rate of the

HOTT gasket is slower, as demonstrated before.

From this comparison between ARLA and hot tightness test results it appears that an evaluation of gasket performance at elevated temperature can be achieved by using the ARLA approach.

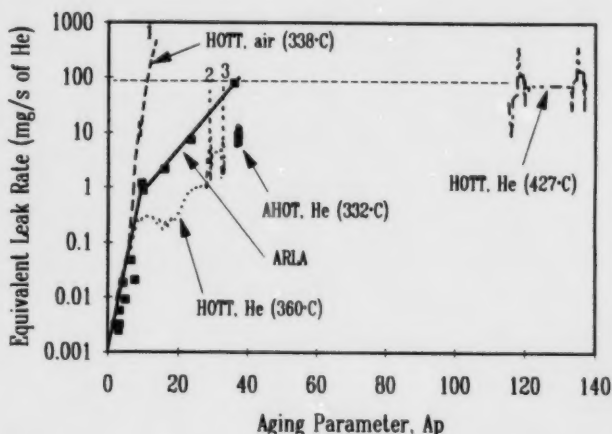


Fig. 8 Effect of internal gas media on gasket leak rate. Comparison between AHOT, HOTT and ARLA test results. NBR-Aramid (N1) sheet material.

CHARACTERIZING GASKET PERFORMANCE WITH A TEMPERATURE INDEX

The evaluation of elevated temperature gasket performance based on weight loss permits the prediction of the maximum operating temperature as a function of service life through a simple three step method as follows:

- 1) Establish the maximum tolerable damage the gasket can sustain while maintaining mechanical integrity and acceptable tightness. In the present case, this is the maximum tolerable weight loss ($w_{max. tol.}$).
- 2) Determine the thermal exposure ($A_{p max.}$) required to obtain $w_{max. tol.}$.
- 3) Use the aging parameter equations (1) or (2) to obtain the maximum operating temperature as a function of the desired service life.

To illustrate how this method works, the maximum operating temperature of the NBR-Aramid sheet gasket (N5) will be evaluated.

Using the approach proposed in the literature [8], $w_{max. tol.}$ for the NBR-Aramid sheet (N5) is found to be between 10% to 15%. By referring to Fig. 4, the corresponding values of $A_{p max.}$ are 17 and 25. Equation (1) can now be rearranged to calculate the maximum operating temperature as a function of $A_{p max.}$ and the service life:

$$T_{op. max.} = \frac{1}{\left(\frac{1}{T_0'} - \frac{A_p max.}{1000 t^{0.6}} \right)} \quad (T_{op. max.} < 330^\circ\text{C}) \quad (3)$$

In Fig. 9, two curves of $T_{op. max.}$ are plotted as a function of service life in a thermal endurance graph. One curve corresponds to a weight loss of 10% while the second is for 15%. From this figure, the maximum operating temperature of the NBR-Aramid sheet (N5) is estimated to be between 193°C and 197°C for a 3-year life. It is interesting to note that for long service life the predicted operating temperatures are not very sensitive to the variation in the maximum tolerable weight loss values (10% vs 15%).

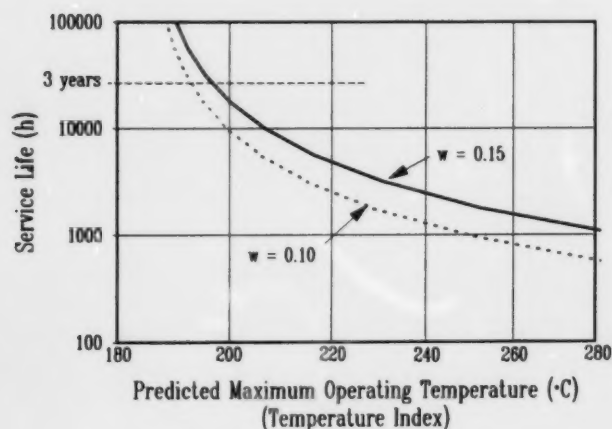


Fig. 9 Predicted maximum operating temperature as a function of service life. NBR-Aramid (N5) sheet material.

The $T_{op. max.}$ calculated with the above method are only exact if the gasket operating conditions are close to the experimental conditions used to determine A_p (ARLA tests). As demonstrated previously with some of the parameters influencing the degradation rate of a sheet gasket, the flange operating conditions will modify the expected service life at the predicted $T_{op. max.}$. For instance, an NBR-Aramid gasket (N5) subjected to an initial compression stress greater than 34.5 MPa (the initial compression stress applied in ARLA test) should have a service life exceeding 3 years at 195°C. On the other hand, the same gasket used at 195°C with pressurized air will have a shorter life expectancy.

It becomes obvious that the predicted $T_{op. max.}$ for a gasket cannot be used directly without taking into account the particular operating conditions of each flange. With this in mind, $T_{op. max.}$ should be considered a temperature index that characterized the elevated temperature performance of a gasket as measured under experimental conditions that are representative of flange operating conditions.

CONCLUSIONS

The recent developments in the evaluation of elevated temperature performance of fiber reinforced sheet gasket materials lead to the following results:

- Strong correlations are shown to exist between gasket weight loss and change in key gasket properties during thermal aging,
- Development of a new aging parameter (A_p) based on gasket weight loss that successfully combines time and temperature effects,
- Parameters such as gasket width, initial compression stress and internal gas medium are shown to influence the thermal degradation rate of gasket,
- A simple method for characterizing elevated temperature gasket performance (ARLA) with a temperature index is shown to be effective.

ACKNOWLEDGMENTS

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A NON-LINEAR THERMOELASTIC SOLID WITH UNIAXIAL WAVE PROPAGATION

M.C. SINGH, D.V.D. TRAN*

Department of Mechanical Engineering
The University of Calgary
Calgary, Alberta, Canada

1. INTRODUCTION

The literature on wave propagation in thermoelastic solids is extensive. After World War II, thermomechanical waves have played an important role to fulfil the requirements of high technology for designs of steam and gas turbines, jet motors, rockets, high-speed aircrafts, nuclear reactors and so on [1]. The earliest work regarding the effects of temperature in the formulation of elasticity problems is that of Duhamel-Newman on the foundation of the constitutive equations of the linear theory of thermoelasticity valid for the general type of anisotropic materials [2]. Based on Duhamel's hypothesis, Danilovskaya took into account inertia effects to investigate the stress distribution in an elastic half space by applying a temperature step function at the boundary [3]. Later, Michaels [4] determined a relation between heating rates and magnitudes of induced stresses in a slender rod when a temperature impact was applied at one end. In their investigation, however, a coupling between thermal and mechanical fields was neglected.

Sneddon [5] is one of the first pioneers who took into account the coupling term in the governing equations. He attempted to calculate the distribution of temperature and stress in a linear thermoelastic thin rod subjected to a mechanical or thermal disturbance at the boundary. Dillon [6], later, studied the dynamic response in a linear thermoelastic thin semi-infinite rod with its near end subjected to either mechanical or thermal disturbance. He also took into account a thermomechanical coupling and the results obtained showed a remarkable deviation from the uncoupled solutions. With the inclusion of coupling in the basic equations, the problem becomes more complicated as the system of basic equations must be solved simultaneously. In practice, the coupling term is usually neglected when its value is very small as compared with unity for the purpose of computational convenience.

In addition to the coupled theory, the classical Fourier's law was also modified by adding a new term to the heat conduction equation to eliminate the paradox of the infinite thermal speed which is not acceptable on the physical grounds. With the aid of modified heat conduction equation, the thermal wave propagates at a finite velocity. Thus, in such a system, there are two different finite waves propagating in the medium, namely the mechanical wave and the thermal wave. A comparison between the classical and modified theories can be seen more clearly in Lopez and Lord's work [7], in which they studied the dynamic response of a thermoelastic half space under various types of boundary conditions. The graphical results they obtained from two theories emphasized the role of the modified Fourier law as a significant contribution to the physics of wave propagation in thermoelastic solids. More recently, Chandrasekharaiah [8] made a significant bibliographical review of the relevant literature dealing with the finite speed of propagation of thermal waves during the past twenty years.

In their investigation of thermomechanical waves, the above authors have focused almost entirely on the linear thermoelastic materials. Within the framework of linearized theory, second order terms in stress-strain relations may be neglected for most of the practical engineering materials. However, some materials such as cast iron or rubber behave nonlinearly. In such materials, the second order terms make significant contributions to the characteristic features of thermoelastic waves. Because of this it is important to take into account their nonlinear behaviour in the constitutive relations.

This study devotes to the examination of wave motion in nonlinear thermoelastic solids. For this purpose, an attempt is made to develop a new constitutive relation for such a solid. The development makes use of the concepts from principles of continuum thermo-mechanics and takes into account Gibbs free energy. On the basis of the nonlinear constitutive relations so developed the fundamental equations of wave propagation for a nonlinear thermoelastic solid have been constructed. Those are further simplified for a nonconducting

* Combustion Dynamics Ltd., Medicine Hat, Alberta, Canada

nonlinear elastic uniaxial material. The initial conditions and boundary conditions are stated. Simple waves and/or shock waves propagating in such a material are also examined.

2. THERMODYNAMIC RELATIONS OF STATE VARIABLES

According to the first law of thermodynamics, the local balance of energy is expressible as [9]:

$$\rho \frac{De}{Dt} = \bar{\sigma} \cdot \frac{D\bar{\epsilon}}{Dt} - \text{div } \bar{q} + \rho r, \quad (1)$$

where e is the internal energy per unit mass, r is the internal heat supply per unit time per unit mass, $\bar{q}$ is the heat flux vector, ρ is mass per unit volume, $\bar{\sigma}$ is the symmetric stress tensor and $\bar{\epsilon}$ is the strain tensor.

In addition, in accordance with the second law, the actual process of thermoelastic deformation of a body subjected to the external forces and non-uniform heating can be expressed by the Clausius-Duhem inequality [9],

$$\rho T \frac{D\mathfrak{A}}{Dt} \geq \rho r - \text{div } \bar{q} + \frac{\bar{q}}{T} \cdot \bar{g}, \quad (2)$$

where

$$\bar{g} = \text{grad } T, \quad (3)$$

and $\mathfrak{A}$ is internal entropy production per unit mass.

Eliminating the term ρr in equation (2) by using equation (1), we obtain:

$$\rho \dot{e} - \rho T \frac{D\mathfrak{A}}{Dt} - \bar{\sigma} \cdot \frac{D\bar{\epsilon}}{Dt} + \frac{\bar{q}}{T} \cdot \bar{g} \leq 0. \quad (4)$$

In this paper, we take the strain $\bar{\epsilon}$ as a state function, whereas the stress $\bar{\sigma}$, temperature T and temperature gradient $\bar{g}$ are taken to be the independent variables, in addition, use is made of the Gibbs free energy G instead of the Helmholtz free energy ψ . The equations of state of a thermoelastic material are assumed in the form:

$$\bar{\epsilon} = \bar{\epsilon} [\bar{\sigma}(\bar{x}, t), T(\bar{x}, t), \bar{g}(\bar{x}, t)], \quad (5a)$$

$$G = \hat{G} [\bar{\sigma}(\bar{x}, t), T(\bar{x}, t), \bar{g}(\bar{x}, t)], \quad (5b)$$

with similar equations for e , $\mathfrak{A}$ and q . In equation (5), G denotes the Gibbs free energy and is defined by [10]:

$$G = e - \mathfrak{A}T - \frac{\bar{\sigma} \cdot \bar{\epsilon}}{\rho}. \quad (6)$$

Taking material derivative on both sides of equation (6) and substituting into equation (1) wherein $\frac{DG}{Dt}$ is eliminated by taking derivative on both sides of equation (5b), we obtain:

$$\left(\rho \frac{\partial \hat{G}}{\partial \bar{\sigma}} + \bar{\epsilon} \right) \cdot \frac{D\bar{\sigma}}{Dt} + \rho \left(\frac{\partial \hat{G}}{\partial T} + \mathfrak{A} \right) \frac{DT}{Dt} + \rho \left(\frac{\partial \hat{G}}{\partial \bar{g}} \right) \cdot \bar{g} \cdot \bar{g} \quad (7)$$

$$\frac{D\bar{g}}{Dt} + \rho \frac{D\mathfrak{A}}{Dt} T + \text{div } \bar{q} - \rho r = 0.$$

Since the expressions appearing in the parentheses in equation (7) do not depend upon $\frac{D\bar{\sigma}}{Dt}$, $\frac{DT}{Dt}$ and $\frac{D\bar{g}}{Dt}$, the expressions must vanish identically. This leads to the consequent conditions:

$$\bar{\epsilon} = -\rho \left(\frac{\partial \hat{G}}{\partial \bar{\sigma}} \right), \quad (8)$$

$$\mathfrak{A} = - \left(\frac{\partial \hat{G}}{\partial T} \right), \quad (9)$$

$$\frac{\partial \hat{G}}{\partial \bar{g}} = 0. \quad (10)$$

Equation (10) confirms that the Gibbs free energy function G is independent of the temperature gradient $\bar{g}$. Since the strain $\bar{\epsilon}$ and the entropy $\mathfrak{A}$ are correlated to G in the terms of stress $\bar{\sigma}$ and temperature T through equations (8) and (9) respectively, we can express strain $\bar{\epsilon}$, internal energy e and entropy $\mathfrak{A}$ as functions which are independent of the temperature gradient $\bar{g}$. It is also noted that only the heat flux $\bar{q}$ is a function of stress $\bar{\sigma}$, temperature T and temperature gradient $\bar{g}$.

The remaining terms in equations (7) and (4) are written as:

$$\rho T \frac{D\mathfrak{A}}{Dt} + \text{div } \bar{q} - \rho r = 0, \quad (11)$$

$$\frac{\bar{q}}{T} \cdot \bar{g} \leq 0. \quad (12)$$

The inequality (12) shows that heat flows only in the direction of decreasing temperature.

Taking total derivative on both sides of equation (9) with respect to time t and a partial derivative on both sides of equation (8) with respect to temperature T , are then combined with equation (11) to obtain:

$$\text{div } \bar{q} - \rho \left(\frac{\partial^2 \hat{G}}{\partial T^2} \right) T + T \frac{\partial \bar{\epsilon}}{\partial T} \cdot \frac{D\bar{\sigma}}{Dt} - \rho r = 0. \quad (13)$$

By definition, the specific heat at constant stress C_σ is given by [11]:

$$C_\sigma = T \left(\frac{\partial \mathfrak{A}}{\partial T} \right)_\sigma = -T \left(\frac{\partial^2 \hat{G}}{\partial T^2} \right)_\sigma \quad (14)$$

Replacing the term $\left(-\frac{\partial^2 \hat{G}}{\partial T^2} \right)$ in equation (13) by C_σ leads to:

$$\text{div } \bar{q} + \rho C_v \frac{DT}{Dt} + T \frac{\partial \bar{\epsilon}}{\partial T} \cdot \frac{D\bar{\sigma}}{Dt} - \rho r = 0. \quad (15)$$

In addition, the modified form of the Fourier's law of heat conduction, usually referred to as Maxwell-Cattaneo equation, is of the form [12]:

$$\tau_v \frac{\partial \bar{q}}{\partial t} + \bar{q} = -K \nabla \theta, \quad (16)$$

where K is the coefficient of thermal conductivity, $\nabla \theta$ is the thermal gradient and τ_v is the thermal relaxation time.

3. STRESS-STRAIN-TEMPERATURE RELATION

Expanding the Gibbs free energy function $\hat{G}$ into a Taylor series in the neighbourhood of its natural state (i.e. when $\bar{\sigma} = 0$, and $T = T_0$), we obtain:

$$\begin{aligned} \hat{G}(\bar{\sigma}, T) = & \hat{G}(0, T_0) + \frac{\partial \hat{G}(0, T_0)}{\partial \bar{\sigma}} \cdot \bar{\sigma} + \\ & \frac{\partial \hat{G}(0, T_0)}{\partial T} \theta + \frac{1}{2} \left[\frac{\partial^2 \hat{G}(0, T_0)}{\partial \bar{\sigma}^2} \cdot \bar{\sigma} \bar{\sigma} + \right. \\ & \left. 2 \frac{\partial^2 \hat{G}(0, T_0)}{\partial \bar{\sigma} \partial T} \cdot \bar{\sigma} \theta + \frac{\partial^2 \hat{G}(0, T_0)}{\partial T^2} \theta^2 \right] + \\ & \frac{1}{6} \left[\frac{\partial^3 \hat{G}(0, T_0)}{\partial \bar{\sigma}^3} \cdot \bar{\sigma} \bar{\sigma} \bar{\sigma} + \right. \\ & \left. 3 \frac{\partial^3 \hat{G}(0, T_0)}{\partial \bar{\sigma}^2 \partial T} \cdot \bar{\sigma} \bar{\sigma} \theta + 3 \frac{\partial^3 \hat{G}(0, T_0)}{\partial \bar{\sigma} \partial T^2} \cdot \bar{\sigma} \theta^2 + \frac{\partial^3 \hat{G}(0, T_0)}{\partial T^3} \theta^3 \right] + \\ & \text{higher order terms,} \end{aligned} \quad (17)$$

where $\theta = T - T_0$ is the temperature difference.

At the natural state, the constant $\hat{G}(0, T_0)$ may be assumed to be zero. From the definition of strain given by equation (8), we have:

$$\begin{aligned} \bar{\epsilon}(\bar{\sigma}, T) = & -\rho \left\{ \frac{\partial \hat{G}(0, T_0)}{\partial \bar{\sigma}} + \frac{\partial \hat{G}(0, T_0)}{\partial \bar{\sigma}^2} \cdot \bar{\sigma} + \right. \\ & \frac{1}{2} \frac{\partial^2 \hat{G}(0, T_0)}{\partial \bar{\sigma}^3} \cdot \bar{\sigma} \bar{\sigma} + \frac{\partial^2 \hat{G}(0, T_0)}{\partial \bar{\sigma} \partial T} \theta + \\ & \left. \frac{\partial^2 \hat{G}(0, T_0)}{\partial \bar{\sigma}^2 \partial T} \cdot \bar{\sigma} \theta + \frac{1}{2} \frac{\partial^2 \hat{G}(0, T_0)}{\partial \bar{\sigma} \partial T^2} \theta^2 \right\}. \end{aligned} \quad (18)$$

Based on the assumption that at the reference state $\bar{\sigma} = \theta = \bar{\epsilon} = 0$, equation (18) can be written in the following form:

$$\bar{\epsilon}(\bar{\sigma}, T) = \bar{a}_1 \cdot \bar{\sigma} + \bar{a}_2 \cdot \bar{\sigma} \bar{\sigma} + \bar{\alpha} \theta, \quad (19)$$

where $\bar{a}_1$ and $\bar{a}_2$ are elastic constant tensors. The last term in equation (19) is considered as a thermal strain tensor.

The above equation expresses the constitutive relation of stress-strain-temperature of a nonlinear thermoelastic material in three dimensions. The equation is considered as a new model of such a material. When the tensor constant $\bar{a}_2$ equals zero, it is apparent that the model reduces to a linear one as commonly used in the literature.

4. BASIC EQUATIONS OF 1-D WAVE PROPAGATION IN NONLINEAR THERMOELASTIC MATERIALS

In one dimension, equation (19) is written as:

$$\epsilon(\sigma, T) = a_2 \sigma + a_3 \sigma^2 + \alpha \theta, \quad (20)$$

where ϵ and σ represent normal strain and normal stress, respectively, a_2 and a_3 are elastic constants and α is the coefficient of linear thermal expansion for isotropic materials.

Equation (20) has an equivalent form such that:

$$\epsilon(\sigma, T) = \left(\frac{\sigma}{\mu} \right)^n + \alpha \theta, \quad (21)$$

where μ is the modulus of elasticity and n is the parameter of nonlinearity. When n equals unity, a linear relation is obtained.

The kinematic compatibility condition is given by [2]:

$$\frac{\partial \epsilon}{\partial t} = \frac{\partial v}{\partial x}, \quad (22)$$

where v is the particle velocity.

The equations (21) and (22) are then combined to yield:

$$\frac{\partial v}{\partial x} = \frac{\partial}{\partial t} \left\{ \left(\frac{\sigma}{\mu} \right)^n + \alpha \theta \right\}. \quad (23)$$

Differentiating equation (21) with respect to temperature T we have:

$$\frac{\partial \epsilon}{\partial T} = \frac{\partial \epsilon}{\partial \theta} = \alpha. \quad (24)$$

With the aid of equation (24), equation (15) can be written in one-dimensional form as:

$$\frac{\partial q}{\partial x} + \rho C_v \frac{\partial \theta}{\partial t} + \alpha T_0 \frac{\partial \sigma}{\partial t} = 0, \quad (25)$$

where the internal heat supply r has been neglected.

In general, the system of equations necessary to investigate one-dimensional wave motion in nonlinear thermoelastic materials is summarized as follows [13]:

Equation of Motion:

$$\frac{\partial \sigma}{\partial x} = \rho \frac{\partial v}{\partial t}, \quad (26a)$$

Constitutive Relation:

$$\frac{\partial v}{\partial x} = \frac{\partial}{\partial t} \left\{ \left(\frac{\sigma}{\mu} \right)^n + \alpha \theta \right\}, \quad (26b)$$

Modified Heat Conduction:

$$\tau_0 \frac{\partial q}{\partial t} + q = -K \frac{\partial \theta}{\partial x}, \quad (26c)$$

Conservation of Energy Equation:

$$\frac{\partial q}{\partial x} + \rho C_v \frac{\partial \theta}{\partial t} + \alpha T_0 \frac{\partial \sigma}{\partial t} = 0. \quad (26d)$$

5. WAVE PROPAGATION IN UNIAXIAL NON-CONDUCTING NONLINEAR ELASTIC SOLIDS

As a special case, in this section we will examine wave propagation in a non-conductor. The concept of non-conducting elastic solids is thermodynamically inadmissible because it is impossible to heat or cool such solids reversibly. In these solids, values of thermal conductivity K equals zero. However, in practice, values of K can never be exactly zero for any solid, because heat can only be transferred through the process of heat conduction. Without heat conduction there would be no reversible path between two states of the solid which differ only in temperature. In a limiting sense, when the value of K is very small, it is assumed that heat flux q in the body is small such that:

$$q = 0. \quad (27)$$

With the aid of equation (27), the fundamental equations of wave propagating in a one-dimensional non-conducting body are obtained from the set of equations (26) as [14]:

$$\rho \frac{\partial v}{\partial t} - \frac{\partial \sigma}{\partial x} = 0, \quad (28a)$$

equation of motion;

$$\frac{\partial}{\partial t} \left\{ \left(\frac{\sigma}{\mu} \right)^n + \alpha \theta \right\} - \frac{\partial v}{\partial x} = 0, \quad (28b)$$

compatibility equation;

$$\alpha T_0 \frac{\partial \sigma}{\partial t} + \rho C_v \frac{\partial \theta}{\partial t} = 0, \quad (28c)$$

conservation of energy;

subjected to the auxiliary conditions:

$$v(0, t) = V_0^*, \quad t > 0, \quad \delta > 0, \quad (29a)$$

$$v(x, 0) = 0, \quad \sigma(x, 0) = 0, \quad \theta(x, 0) = 0; \quad x > 0. \quad (29b, c, d)$$

In equation (29a) V_0 and δ are parameters of the time-dependent velocity impact. Tensile stress is assumed to be positive.

After rearranging, the system (28) can be written in a general form as:

$$U_t + [B]U_x = 0, \quad (30)$$

where

$$U_t = \frac{\partial U}{\partial t} = [v, \sigma, \theta]^T, \quad (31a)$$

$$U_x = \frac{\partial U}{\partial x} = [v_x, \sigma_x, \theta_x]^T, \quad (31b)$$

and the matrix $[B]$ is:

$$[B] = \begin{bmatrix} 0 & \frac{1}{\rho} & 0 \\ -\rho C_v & 0 & 0 \\ \frac{\alpha T_0}{(\beta \rho C_v - \alpha^2 T_0)} & 0 & 0 \end{bmatrix}. \quad (31c)$$

In the above equations, superscript 'T' denotes the transpose of a matrix, and β is a function of stress defined by:

$$\beta = \frac{n}{\mu^n} \sigma^{n-1}. \quad (32)$$

The characteristic relation of equation (30) is given by:

$$|B - \lambda I| = 0. \quad (33)$$

The eigenvalues λ_i , $i = 1, 2, 3$ of equation (33) are determined as:

$$\lambda_1 = 0, \quad (34a)$$

$$\lambda_2 = \sqrt{\frac{C_v}{\beta \rho C_v - \alpha^2 T_0}}, \quad (34b)$$

$$\lambda_3 = -\sqrt{\frac{C_v}{\beta \rho C_v - \alpha^2 T_0}}. \quad (34c)$$

By definition, the wave speeds along the characteristic curves are expressible as:

$$\text{-along } C_1: \quad V_1 = \lambda_1 = 0, \quad (35a)$$

$$\text{-along } C_2^+: \quad V_2 = \frac{dx}{dt} = \lambda_2, \quad (35b)$$

$$\text{-along } C_2^-: \quad V_3 = -\frac{dx}{dt} = \lambda_3. \quad (35c)$$

The equations along the characteristic curves found from the left eigenvectors associated with the matrix $[B]$ are given by:

$$\text{-along } C_1: \quad \frac{d\sigma}{dt} + \rho \frac{C_\sigma}{\alpha T_\sigma} \frac{d\theta}{dt} = 0, \quad (36a)$$

$$\text{-along } C_2^+: \quad -\rho V_2 \frac{dv}{dt} + \frac{d\sigma}{dt} = 0, \quad (36b)$$

$$\text{-along } C_2^-: \quad \rho V_2 \frac{dv}{dt} + \frac{d\sigma}{dt} = 0. \quad (36c)$$

By observing the positive wave speeds expressed by equation (34b) which is a function of stress σ , the family of characteristic curves either converge or diverge at the front depending on the type of impact applied to the problem. Simple waves or shock waves, thus, may occur in the material during the time of wave propagation [15].

5.1 Simple Waves: In the case of simple waves, the wave front is explicitly determined from the characteristic curve C_2^+ passing through the origin and jumps in unknowns across the wave front. According to the Kosinski's theorem [16], the jump condition associated with equation (28a) is expressible as:

$$|\sigma| = -\rho V_2 |v| \quad (37a)$$

On the basis of equation (28c), the jump condition is:

$$|\sigma| = -\frac{\rho C_\sigma}{\alpha T_\sigma} |\theta|, \quad (37b)$$

where ' | ' denotes the jump.

5.2 Shock Waves: Unlike the simple waves, the problem becomes more complicated when there are shock waves involved. To analyze the path of shock front as well as the unknown values of the jump, the following set of equations are obtained on the basis of equations of kinematic compatibility [17] and the fundamental equations (28) at the front:

$$|\sigma| = -\rho U_s |v| \quad (38a)$$

$$|\sigma| = -\frac{C_\sigma}{\alpha T_\sigma} |\theta|, \quad (38b)$$

$$2U_s \frac{d|\epsilon|}{dt} + |\epsilon| \frac{dU_s}{dt} = U_s^2 \left| \frac{\partial \epsilon}{\partial x} \right| - \frac{1}{\rho} \left| \frac{\partial \sigma}{\partial x} \right|, \quad (37c)$$

$$U_s = \sqrt{\frac{|\sigma|}{\rho |\epsilon|}}, \quad (37d)$$

$$|\epsilon| = \left| \left(\frac{\sigma}{v} \right) \right| + \alpha |\theta|, \quad (37e)$$

where U_s is the intrinsic velocity of the shock front. The equation (37c) is also recognized as the shock amplitude equation [17].

6. CONCLUSIONS

The new constitutive equation for a class of nonlinear thermoelastic materials is developed on the basis of modern continuum thermomechanics and the Gibbs free energy.

Fundamental equations expressing wave propagation in such a material are essentially based on the physical nonlinearity of the material represented by the constitutive relation (19) or (21). The value of power n in equation (21) varies with each material and is usually greater than unity for metals. When the value of n equal unity, the linear relation of thermoelastic materials [5, 6] is obtained as a special case.

Uniaxial wave propagation in a nonlinear thermoelastic nonconducting material is investigated. In such a material, heat flux is assumed to be absent and only one wave front propagates. Under a loading condition ($\delta > 0$) at the boundary, there is always a simple wave propagating in the material having a value of n greater than unity and there is a shock wave propagating in the material having n less than unity. As shown in Fig. 1, the distribution of stresses along a string of rubber-like material ($n = 1.6$) reveals that the stress jump is constant. Whereas, in a string of leather-like material ($n = 0.7545$), the stress jump is increasing at the shock front as it propagates (Fig. 2). Since the shock occurs, the growth of shock amplitude leads to the growth of stress along the shock path. This physical aspect is important in dynamic designs as the continuous growth of stress with time may ultimately and indefinitely cause the material failure which is not expected in the case of linear elastic materials.

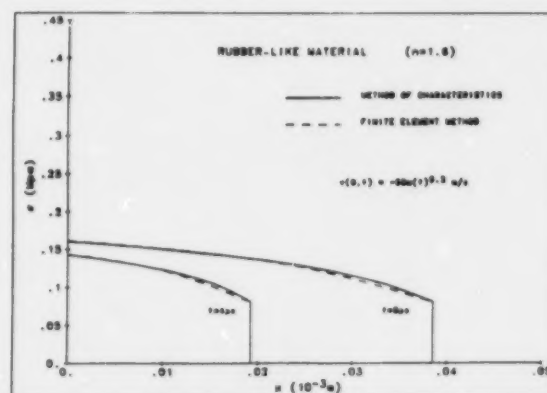


Fig. 1 - Stress Response of a Rubber-Like Material to Velocity Impact.

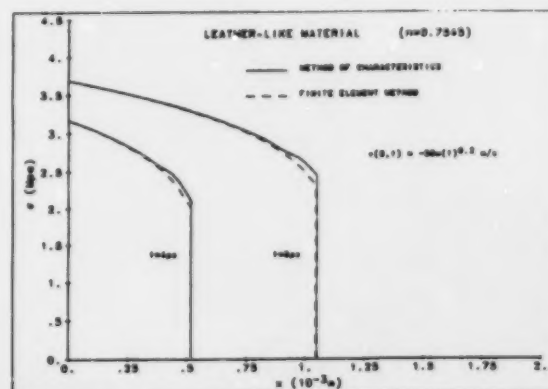


Fig. 2 - Stress Response of a Leather-Like Material to Velocity Impact.

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BENDING ANALYSIS OF ORTHOTROPIC CONICAL SHELLS

Liyong Tong and B. Tabarrok

Department of Mechanical Engineering
University of Victoria, Victoria, B.C., Canada V8W 3P6

Tsun Kuei Wang

Department of Applied Mechanics
Beijing University of Aeronautics and Astronautics
Beijing, P.R. China

ABSTRACT

Using Donnell type shell theory a simple and exact procedure is presented for bending analysis of orthotropic conical shells under various loads. The solution is in the form of a power series in terms of a particularly convenient coordinate system, and its convergence radius is obtained.

INTRODUCTION

Orthotropic Conical shells, manufactured from composite materials or concrete materials or with stiffeners, find many applications in certain fields of engineering, such as aeronautical and aerospace, chemical and nuclear engineering, etc. These applications include these uses of conical shells as the conjunction part between two cylindrical shells with different diameters, as nuclear reactor walls, as antenna etc. These structures are often subjected to different loads, including thermal and moisture expansions. To assess some aspects of the mechanical performance of such shells, it is important to execute a static bending analysis. Much literatures exists on the static bending analysis of isotropic conical shells. Watts and Lang (1952) studied the axisymmetrical bending behaviour of a complete conical shell used as a cap for a cylindrical pressure vessel. Huth (1953) investigated the thermal stresses in conical shells. Taylor and Wenk (1955) and Baltrukonis (1959) considered edge effects in long and short conical shells, respectively. Using Donnell type theory, Seide (1957) derived the governing equation for asymmetrical bending of thin conical shells. For the case of orthotropic conical shells, fewer studies exist. Singer (1963) developed the Donnell type equations for bending and buckling analysis of orthotropic shells. In the present investigation, a simple and exact procedure for bending analysis of isotropic and orthotropic conical shells under certain types of loads is developed. The procedure consists of the following steps:

- the bending equations are developed and expressed in terms of displacements;

- using a new technique, exact solutions are constructed in series form for the governing equations;
- convergence properties of the series solution are determined.

DONNELL-TYPE GOVERNING EQUATIONS

Consider a conical shell as shown in Fig. 1. R_1 and R_2 indicate the radii of the cone at its small and large ends, respectively, α is the semivertex angle of the cone and L is the cone length along its generator. We now introduce the $x - \phi$ coordinate system; x is measured along the cone's generator starting at middle length and ϕ is the circumferential coordinate. The displacements of the shell's middle surface are denoted by U and V along x and ϕ directions respectively, and by W along the normal to the surface (inward positive). In terms of these variables the cone's radius at any point along its length may be expressed as

$$R(x) = R_0 + x \sin \alpha \quad (1)$$

For linear bending analysis of orthotropic conical shells, under an external pressure q (q is a function of x and ϕ), we adopt the shallow shell theory of Donnell-type and write the governing equations derived in Tong (1988) as

$$\begin{aligned} L_{11}U + L_{12}V + L_{13}W &= 0 \\ L_{21}U + L_{22}V + L_{23}W &= 0 \\ L_{31}U + L_{32}V + L_{33}W &= q \end{aligned} \quad (2)$$

where

$$\begin{aligned} L_{11} &= A_{11} \frac{\partial^2}{\partial x^2} \\ &\quad + \frac{A_{11} \sin \alpha}{R(x)} \frac{\partial}{\partial x} - \frac{A_{22} \sin^2 \alpha}{R^2(x)} + \frac{A_{66}}{R^2(x)} \frac{\partial^2}{\partial \phi^2} \\ L_{12} &= \frac{(A_{12} + A_{66})}{R(x)} \frac{\partial^2}{\partial x \partial \phi} - \frac{(A_{22} + A_{66}) \sin \alpha}{R^2(x)} \frac{\partial}{\partial \phi} \end{aligned}$$

$$\begin{aligned}
L_{21} &= \frac{(A_{12} + A_{66})}{R(x)} \frac{\partial^2}{\partial x \partial \phi} + \frac{(A_{22} + A_{66}) \sin \alpha}{R^2(x)} \frac{\partial}{\partial \phi} \\
L_{22} &= A_{66} \left[\frac{\partial^2}{\partial x^2} + \frac{\sin \alpha}{R(x)} \frac{\partial}{\partial x} - \frac{\sin^2 \alpha}{R^2(x)} \right] + \frac{A_{22}}{R^2(x)} \frac{\partial^2}{\partial \phi^2} \\
L_{13} &= -\frac{A_{12} \cos \alpha}{R(x)} \frac{\partial}{\partial x} + \frac{A_{22} \sin \alpha \cos \alpha}{R^2(x)} \\
L_{23} &= L_{32} = -\frac{A_{22} \cos \alpha}{R^2(x)} \frac{\partial}{\partial \phi} \\
L_{31} &= -\frac{A_{12} \cos \alpha}{R(x)} \frac{\partial}{\partial x} - \frac{A_{22} \sin \alpha \cos \alpha}{R^2(x)} \\
L_{33} &= \frac{A_{22} \cos^2 \alpha}{R^2(x)} + D_{11} \frac{\partial^4}{\partial x^4} + \frac{2(D_{12} + 2D_{66})}{R^2(x)} \frac{\partial^4}{\partial x^2 \partial \phi^2} \\
&\quad + \frac{D_{22}}{R^4(x)} \frac{\partial^4}{\partial \phi^4} + \frac{2D_{11} \sin \alpha}{R(x)} \frac{\partial^3}{\partial x^3} \\
&\quad - \frac{2(D_{12} + 2D_{66}) \sin \alpha}{R^3(x)} \frac{\partial^3}{\partial x \partial \phi^2} - \frac{D_{22} \sin^2 \alpha}{R^2(x)} \frac{\partial^2}{\partial x^2} \\
&\quad + \frac{2(D_{12} + D_{22} + 2D_{66}) \sin^2 \alpha}{R^4(x)} \frac{\partial^2}{\partial \phi^2} \\
&\quad + \frac{D_{22} \sin^3 \alpha}{R^3(x)} \frac{\partial}{\partial x}
\end{aligned} \quad (3)$$

and A_{ij} and D_{ij} ($i, j=1,2,6$) are calculated from following equations

$$\begin{aligned}
A_{11} &= \frac{E_x h}{1 - \mu_{x\phi} \mu_{\phi x}} & A_{12} &= \frac{\mu_{\phi x} E_x h}{1 - \mu_{x\phi} \mu_{\phi x}} \\
A_{21} &= \frac{\mu_{x\phi} E_\phi h}{1 - \mu_{x\phi} \mu_{\phi x}} & A_{22} &= \frac{E_\phi h}{1 - \mu_{x\phi} \mu_{\phi x}} \\
A_{66} &= G_{x\phi} h \\
D_{11} &= \frac{E_x h^3}{12(1 - \mu_{x\phi} \mu_{\phi x})} & D_{12} &= \frac{\mu_{\phi x} E_x h^3}{12(1 - \mu_{x\phi} \mu_{\phi x})} \\
D_{21} &= \frac{\mu_{x\phi} E_\phi h^3}{12(1 - \mu_{x\phi} \mu_{\phi x})} & D_{22} &= \frac{E_\phi h^3}{12(1 - \mu_{x\phi} \mu_{\phi x})} \\
D_{66} &= \frac{G_{x\phi} h^3}{12}
\end{aligned} \quad (4)$$

in which E_x , E_ϕ , $\mu_{x\phi}$, $\mu_{\phi x}$ and $G_{x\phi}$ are material elastic constants; h is the wall thickness and

$$\mu_{x\phi} E_\phi = \mu_{\phi x} E_x$$

The force and moment stress resultants are expressed in terms of displacements U , V and W by

$$\begin{Bmatrix} N_x \\ N_\phi \\ N_{x\phi} \\ M_x \\ M_\phi \\ M_{x\phi} \end{Bmatrix} = \begin{bmatrix} l_{11} & l_{12} & l_{13} \\ l_{21} & l_{22} & l_{23} \\ l_{31} & l_{32} & 0 \\ 0 & 0 & l_{43} \\ 0 & 0 & l_{53} \\ 0 & 0 & l_{63} \end{bmatrix} \begin{Bmatrix} U \\ V \\ W \end{Bmatrix} \quad (5)$$

where

$$l_{11} = A_{11} \frac{\partial}{\partial x} + \frac{A_{12} \sin \alpha}{R(x)}$$

$$\begin{aligned}
l_{12} &= \frac{A_{12}}{R(x)} \frac{\partial}{\partial \phi} \\
l_{13} &= -\frac{A_{12} \cos \alpha}{R(x)} \\
l_{31} &= \frac{A_{66}}{R(x)} \frac{\partial}{\partial \phi} \\
l_{32} &= A_{66} \left(\frac{\partial}{\partial x} - \frac{\sin \alpha}{R(x)} \right) \\
l_{13} &= -D_{11} \frac{\partial^2}{\partial x^2} - \frac{D_{12} \sin \alpha}{R(x)} \frac{\partial}{\partial x} - \frac{D_{12}}{R^2(x)} \frac{\partial^2}{\partial \phi^2} \\
l_{63} &= -D_{66} \frac{\partial}{\partial x} \left[\frac{1}{R(x)} \frac{\partial}{\partial \phi} \right]
\end{aligned} \quad (6)$$

where $i = 1, 2$ and $j = 3 + i$.

The transverse shear force resultants can be obtained from M_x , M_ϕ and $M_{x\phi}$ by

$$\begin{aligned}
Q_x &= \frac{1}{R(x)} \frac{\partial}{\partial x} [R(x) M_x] - \frac{M_\phi \sin \alpha}{R(x)} + \frac{1}{R(x)} \frac{\partial M_{x\phi}}{\partial \phi} \\
Q_\phi &= \frac{1}{R(x)} \frac{\partial}{\partial x} [R(x) M_\phi] + \frac{M_{x\phi} \sin \alpha}{R(x)} + \frac{1}{R(x)} \frac{\partial M_x}{\partial \phi}
\end{aligned} \quad (7)$$

The related boundary conditions may be expressed as

$$\begin{aligned}
N_x &= \bar{N}_x \quad \text{or} \quad U = \bar{U} \\
N_{x\phi} &= \bar{N}_{x\phi} \quad \text{or} \quad V = \bar{V} \\
M_x &= \bar{M}_x \quad \text{or} \quad \frac{\partial W}{\partial x} = \frac{\partial \bar{W}}{\partial x} \quad \text{when } x = \pm \frac{L}{2} \\
Q_x &= \bar{Q}_x \quad \text{or} \quad W = \bar{W}
\end{aligned} \quad (8)$$

The above set of governing equations degenerate to those of cylindrical shells when α is set equal to zero. It is also worth noting that if the starting point of the x axis is changed to the cone's vertex, where the radius is equal to zero, the $x - \phi$ coordinate system will coincide with the $s - \phi$ coordinate system used by many previous researchers. A further point of interest is the case when α , the semivertex angle, approaches a right angle. In this case, the differential operators L_{13} , L_{23} , L_{31} and L_{32} approach zero and the three equilibrium equations become independent, that is, the first two equations will then describe the in-plane problem and the third the bending problem of circular plates under transverse load.

Evidently the system of governing equations presented in the foregoing is complex and to our knowledge exact solutions have not been given for these equations. In the following section we outline a strategy for constructing general solutions for these equations.

EXACT SOLUTIONS

An inspection of the differential operators L_{ij} ($i, j = 1, 2, 3$) in Eqs (3) reveals the following properties:

The coefficients of all these operators are functions of x only, i.e., they are independent of ϕ , and they include terms of the following form: $1/R^k(x)$, $k = 0, 1, 2, 3, 4$. For the operators L_{ij} ($i = 1, 2; j = 1, 2, 3$), k takes values from zero to two. For L_{3j} ($j = 1, 2, 3$) the value of k ranges from zero to four. These useful properties allow us to change the equations into a more

convenient form. Multiplying the first two equations of (3) by $R^2(x)$ and the third by $R^4(x)$ we obtain the following modified equations.

$$\begin{aligned} L_{11}^* U + L_{12}^* V + L_{13}^* W &= 0 \\ L_{21}^* U + L_{22}^* V + L_{23}^* W &= 0 \\ L_{31}^* U + L_{32}^* V + L_{33}^* W &= R^4(x) q \end{aligned} \quad (9)$$

where

$$\begin{aligned} L_{11}^* &= A_{11} R^2(x) \frac{\partial^2}{\partial x^2} + A_{11} R(x) \sin \alpha \frac{\partial}{\partial x} - A_{22} \sin^2 \alpha + A_{66} \frac{\partial^2}{\partial \phi^2} \\ L_{12}^* &= (A_{12} + A_{66}) R(x) \frac{\partial^2}{\partial x \partial \phi} - (A_{22} + A_{66}) \sin \alpha \frac{\partial}{\partial \phi} \\ L_{21}^* &= (A_{12} + A_{66}) R(x) \frac{\partial^2}{\partial x \partial \phi} + (A_{22} + A_{66}) \sin \alpha \frac{\partial}{\partial \phi} \\ L_{22}^* &= A_{66} [R^2(x) \frac{\partial^2}{\partial x^2} + R(x) \sin \alpha \frac{\partial}{\partial x} - \sin^2 \alpha] + A_{22} \frac{\partial^2}{\partial \phi^2} \\ L_{13}^* &= -A_{12} R(x) \cos \alpha \frac{\partial}{\partial x} + A_{22} \sin \alpha \cos \alpha \\ L_{23}^* &= -A_{22} \cos \alpha \frac{\partial}{\partial \phi} \\ L_{31}^* &= -R^2(x) [A_{12} R(x) \cos \alpha \frac{\partial}{\partial x} - A_{22} \sin \alpha \cos \alpha] \\ L_{32}^* &= -A_{22} R^2(x) \cos \alpha \frac{\partial}{\partial \phi} \\ L_{33}^* &= D_{11} R^4(x) \frac{\partial^4}{\partial x^4} + 2(D_{12} + 2D_{66}) R^2(x) \frac{\partial^4}{\partial x^2 \partial \phi^2} + D_{22} \frac{\partial^4}{\partial \phi^4} + 2D_{11} R^2(x) \sin \alpha \frac{\partial^3}{\partial x^3} - 2(D_{12} + 2D_{66}) R(x) \sin \alpha \frac{\partial^3}{\partial x \partial \phi^2} - D_{22} R^2(x) \sin^2 \alpha \frac{\partial^2}{\partial x^2} + D_{22} R(x) \sin^3 \alpha \frac{\partial}{\partial x} + 2(D_{12} + D_{22} + 2D_{66}) \sin^2 \alpha \frac{\partial^2}{\partial \phi^2} + A_{22} R^2(x) \cos^2 \alpha \end{aligned} \quad (10)$$

Now let us consider a special type of load given by

$$q = q_0 \cos n \phi \quad (11)$$

which represents only one term in the Fourier series of an arbitrarily distributed load function q . In this case, let us assume solutions for equations (9), of the following form:

$$\begin{aligned} U &= u(x) \cos n \phi \\ V &= v(x) \sin n \phi \\ W &= w(x) \cos n \phi \end{aligned} \quad (12)$$

where

$$u(x) = \sum_{m=0}^{\infty} a_m x^m$$

$$\begin{aligned} v(x) &= \sum_{m=0}^{\infty} b_m x^m \\ w(x) &= \sum_{m=0}^{\infty} c_m x^m \end{aligned} \quad (13)$$

and n is an integer representing the circumferential wave number of the buckled shell; a_m , b_m and c_m are constants to be determined later.

On substituting from equations (12) and (13) into equation (9) and using equations (1) and (10) we develop three linear algebraic equations by matching the terms of same order in x , and in addition we obtain the following recurrence relations

$$\begin{aligned} a_{m+2} &= G_{1,1} a_{m+1} + G_{1,2} a_m + G_{1,3} b_{m+1} + G_{1,4} b_m \\ &\quad + G_{1,5} c_{m+1} + G_{1,6} c_m \\ b_{m+2} &= G_{2,1} a_{m+1} + G_{2,2} a_m + G_{2,3} b_{m+1} + G_{2,4} b_m \\ &\quad + G_{2,5} c_m \\ c_{m+4} &= G_{3,1} a_{m+1} + G_{3,2} a_m + G_{3,3} a_{m-1} + G_{3,4} a_{m-2} \\ &\quad + G_{3,5} b_m + G_{3,6} b_{m-1} + G_{3,7} b_{m-2} + G_{3,8} c_{m+3} \\ &\quad + G_{3,9} c_{m+2} + G_{3,10} c_{m+1} + G_{3,11} c_m + G_{3,12} c_{m-1} \\ &\quad + G_{3,13} c_{m-2} + F_m \end{aligned} \quad (m = 0, 1, 2, \dots) \quad (14)$$

where the coefficients F_m ($m = 0, 1, 2, \dots$) and $G_{i,j}$ ($(i, j) = (1, 6), (2, 5)$ and $(3, 13)$) are given in the Appendix. The above recurrence relations allow one to express the unknown constants a_m , b_m ($m \geq 2$) and c_m ($m \geq 4$) in terms of $a_0, a_1, b_0, b_1, c_0, c_1, c_2$ and c_3 . Therefore the general form of $u(x)$, $v(x)$ and $w(x)$ may be written as

$$\begin{aligned} u(x) &= u_0(x) + u_1(x) a_0 + u_2(x) a_1 + u_3(x) b_0 \\ &\quad + u_4(x) b_1 + u_5(x) c_0 + u_6(x) c_1 + u_7(x) c_2 \\ &\quad + u_8(x) c_3 \\ v(x) &= v_0(x) + v_1(x) a_0 + v_2(x) a_1 + v_3(x) b_0 \\ &\quad + v_4(x) b_1 + v_5(x) c_0 + v_6(x) c_1 + v_7(x) c_2 \\ &\quad + v_8(x) c_3 \\ w(x) &= w_0(x) + w_1(x) a_0 + w_2(x) a_1 + w_3(x) b_0 \\ &\quad + w_4(x) b_1 + w_5(x) c_0 + w_6(x) c_1 + w_7(x) c_2 \\ &\quad + w_8(x) c_3 \end{aligned} \quad (15)$$

in which $u_0(x)$, $v_0(x)$ and $w_0(x)$ are the particular solution, and $u_i(x)$, $v_i(x)$ and $w_i(x)$ ($i = 1, 2, \dots, 8$) are the base functions of $u(x)$, $v(x)$ and $w(x)$, respectively, and $a_0, a_1, b_0, b_1, c_0, c_1, c_2$ and c_3 are the unknowns to be determined by imposition of the boundary conditions at both ends of the cone.

Before going into details of the solution procedure, let us consider the convergence property of the series solutions $u(x)$, $v(x)$ and $w(x)$ defined in Eqs (13) and the corresponding recurrence Eqs (14).

Careful analysis of the recurrence Eqs (14) and the coefficients F_m and $G_{i,j}$ given in the Appendix shows that:

- The power series defined in Eqs (13) and (14) are alternating series, i.e., the terms of the series change sign consecutively. This property can be readily verified through numerical calculations [7].

- When m becomes large enough, the recurrence equations (14) can be written approximately as follows

$$\begin{aligned} a_{m+2} &\doteq -\frac{2\sin\alpha}{R_0}a_{m+1} - \frac{\sin^2\alpha}{R_0^2}a_m \\ b_{m+2} &\doteq -\frac{2\sin\alpha}{R_0}b_{m+1} - \frac{\sin^2\alpha}{R_0^2}b_m \\ c_{m+4} &\doteq -\frac{4\sin\alpha}{R_0}c_{m+3} - \frac{6\sin^2\alpha}{R_0^2}c_{m+2} \\ &\quad - \frac{4\sin^3\alpha}{R_0^3}c_{m+1} - \frac{\sin^4\alpha}{R_0^4}c_m \end{aligned} \quad (16)$$

These approximate recurrence equations indicate that the coefficients a_m, b_m and c_m are predominantly dependent on the former terms a_i, b_i ($i = 0, 1$) and c_i ($i = 0, 1, 2, 3$), respectively, when m is large enough.

Assuming the convergence ratio of $u(x), v(x)$ and $w(x)$ to be ρ_a, ρ_b and ρ_c , respectively, i.e.,

$$\begin{aligned} \rho_a &= \lim_{m \rightarrow \infty} \frac{a_{m+1}}{a_m} \\ \rho_b &= \lim_{m \rightarrow \infty} \frac{b_{m+1}}{b_m} \\ \rho_c &= \lim_{m \rightarrow \infty} \frac{c_{m+1}}{c_m} \end{aligned} \quad (17)$$

and noting the elementary character of the alternating series, Eqs (16) can be changed into the following form:

$$\begin{aligned} \rho_a^2 &= \frac{2\sin\alpha}{R_0}\rho_a - \frac{\sin^2\alpha}{R_0^2} \\ \rho_b^2 &= \frac{2\sin\alpha}{R_0}\rho_b - \frac{\sin^2\alpha}{R_0^2} \\ \rho_c^4 &= \frac{4\sin\alpha}{R_0}\rho_c^3 - \frac{6\sin^2\alpha}{R_0^2}\rho_c^2 + \frac{4\sin^3\alpha}{R_0^3}\rho_c - \frac{\sin^4\alpha}{R_0^4} \end{aligned} \quad (18)$$

It is now easy to verify, by simple manipulations, that ρ_a and ρ_b have the following identical real roots

$$\rho_a = \rho_b = \rho = \frac{\sin\alpha}{R_0} \quad (19)$$

and ρ_c has a single real root

$$\rho_c = \rho = \frac{\sin\alpha}{R_0} \quad (20)$$

therefore, the series $u(x), v(x)$ and $w(x)$ obtained in the previous part have identical convergence radius r_c , i.e.,

$$r_c = \frac{1}{\rho} = \frac{R_0}{\sin\alpha} \quad (21)$$

Noting that

$$R_0 = \frac{R_1 + R_2}{2} \quad (22)$$

the convergence radius for the three series becomes

$$r_c = \frac{R_1 + R_2}{2\sin\alpha} \quad (23)$$

That is as long as x is within the circle of radius r_c , convergence will be assured. For the shells considered here, the maximum value of $|x|$ is $L/2$. Thus for our purposes, the condition for convergence is

$$\frac{L}{2} \leq \frac{R_1 + R_2}{2\sin\alpha} \quad (24)$$

The convergence condition (26) can be rewritten as

$$L\sin\alpha \leq R_1 + R_2 \quad (25)$$

or

$$R_2 - R_1 \leq R_1 + R_2 \quad (26)$$

This may finally be written as

$$R_1 \geq 0 \quad (27)$$

Hence the three constructed series will converge to their corresponding solutions if the small radius is not zero, i.e., if the conical shell is a truncated one. A complete cone is treated as a truncated cone with a very small radius at its apex. Thus for all practical purposes, there are no limitations on the geometric parameters of the shell considered. Accordingly, the solution obtained provides exact solutions for the three displacements U, V and W for the bending of cones under various types of loads. The three displacements U, V and W may be used to calculate the stress resultants N_x, N_ϕ and $N_{x\phi}$ and the bending moments M_x, M_ϕ and $M_{x\phi}$ through Eqs (5) and (6), and furthermore the transverse shear forces Q_x and Q_ϕ may be obtained from Eqs (7). This solution is exact because it satisfies the governing equations rigorously and it also satisfies the eight boundary conditions through eight arbitrary constants.

CONCLUDING REMARKS

Using the power series method and choosing a particularly convenient coordinate system, a simple solution procedure is developed for bending analysis of isotropic and orthotropic conical shells under certain types of loads. The solution is applicable to all types of boundary conditions, and is particularly useful in the bending analysis of the edge effects.

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APPENDIX

The coefficients F_m in Eqs (14) are given by

$$\begin{aligned} F_0 &= \frac{1}{24 D_{11}} \\ F_1 &= \frac{\sin \alpha}{30 R_o D_{11}} \\ F_2 &= \frac{\sin^2 \alpha}{60 R_o^2 D_{11}} \\ F_3 &= \frac{\sin^3 \alpha}{210 R_o^3 D_{11}} \\ F_4 &= \frac{\sin^4 \alpha}{1680 R_o^4 D_{11}} \\ F_m &= 0 \quad (m = 5, 6, \dots) \end{aligned}$$

The coefficients $G_{i,j}$ in Eqs (14) are expressed as follows

$$\begin{aligned} G_{1,1} &= -\frac{(2m+1)\sin \alpha}{(m+2)R_o} \\ G_{1,2} &= -\frac{m^2 \sin^2 \alpha}{(m+2)(m+1)R_o^2} + \frac{A_{22} \sin^2 \alpha + A_{66} n^2}{A_{11} R_o^2 (m+2)(m+1)} \\ G_{1,3} &= -\frac{(A_{12} + A_{66})n}{A_{11} R_o (m+2)} \\ G_{1,4} &= -\frac{[(A_{12} + A_{66})m - (A_{22} + A_{66})]n \sin \alpha}{A_{11} R_o^2 (m+2)(m+1)} \\ G_{1,5} &= \frac{A_{12} \cos \alpha}{A_{11} R_o (m+2)} \\ G_{1,6} &= \frac{(A_{12}m - A_{22}) \sin \alpha \cos \alpha}{A_{11} R_o^2 (m+2)(m+1)} \\ G_{2,1} &= \frac{(A_{12} + A_{66})n}{A_{66} R_o (m+2)} \\ G_{2,2} &= \frac{[(A_{12} + A_{66})m + (A_{22} + A_{66})]n \sin \alpha}{A_{66} R_o^2 (m+2)(m+1)} \\ G_{2,2} &= G_{1,1} \\ G_{2,4} &= -\frac{m^2 \sin^2 \alpha}{(m+2)(m+1)R_o^2} + \frac{A_{66} \sin^2 \alpha + A_{22} n^2}{A_{66} R_o^2 (m+2)(m+1)} \\ G_{2,5} &= -\frac{A_{22} n \cos \alpha}{A_{66} R_o^2 (m+2)(m+1)} \\ G_{3,1} &= \frac{A_{12} \cos \alpha}{D_{11} R_o (m+4)(m+3)(m+2)} \\ G_{3,2} &= \frac{(3A_{12}m + A_{22}) \sin \alpha \cos \alpha}{D_{11} R_o^2 m_{41}} \\ G_{3,3} &= \frac{[3A_{12}(m-1) + 2A_{22}] \sin^2 \alpha \cos \alpha}{D_{11} R_o^3 m_{41}} \\ G_{3,4} &= \frac{[A_{12}(m-2) + A_{22}] \sin^3 \alpha \cos \alpha}{D_{11} R_o^4 m_{41}} \\ G_{3,5} &= \frac{A_{22} n \cos \alpha}{D_{11} R_o^3 m_{41}} \\ G_{3,6} &= \frac{A_{22} n \sin 2\alpha}{D_{11} R_o^3 m_{41}} \\ G_{3,7} &= \frac{A_{22} n \sin^2 \alpha \cos \alpha}{D_{11} R_o^4 m_{41}} \end{aligned}$$

$$\begin{aligned} G_{3,8} &= -\frac{2(2m+1)\sin \alpha}{R_o(m+4)} \\ G_{3,9} &= \left[\frac{2(D_{12} + 2D_{66})n^2 + D_{22} \sin^2 \alpha}{D_{11} R_o^2} \right. \\ &\quad \left. + \frac{6m^2 \sin^2 \alpha}{R_o^2} \right] / (m+4)(m+3) \\ G_{3,10} &= \left\{ \frac{2m(m-1)(2m-1)\sin^3 \alpha}{R_o^3} + \right. \\ &\quad \left. \frac{[2(D_{12} + 2D_{66})n^2 + D_{22} \sin^2 \alpha](2m-1)\sin \alpha}{D_{11} R_o^3} \right\} \\ &\quad / (m+4)(m+3)(m+2) \\ G_{3,11} &= \{ [2(D_{12} + 2D_{66})n^2 + D_{22} \sin^2 \alpha] \sin^2 \alpha m(m-2) \\ &\quad - A_{22} R_o^2 \cos^2 \alpha - D_{11} m(m-1)^2(m-2) \sin^4 \alpha - D_{22} n^4 \\ &\quad + 2(D_{12} + D_{22} + 2D_{66})n^2 \sin^2 \alpha \} / (D_{11} R_o^4 m_{41}) \\ G_{3,12} &= -2A_{22} R_o \sin \alpha \cos^2 \alpha / (D_{11} R_o^4 m_{41}) \\ G_{3,13} &= -A_{22} \sin^2 \alpha \cos^2 \alpha / (D_{11} R_o^4 m_{41}) \end{aligned}$$

where $m_{41} = (m+4)(m+3)(m+2)(m+1)$

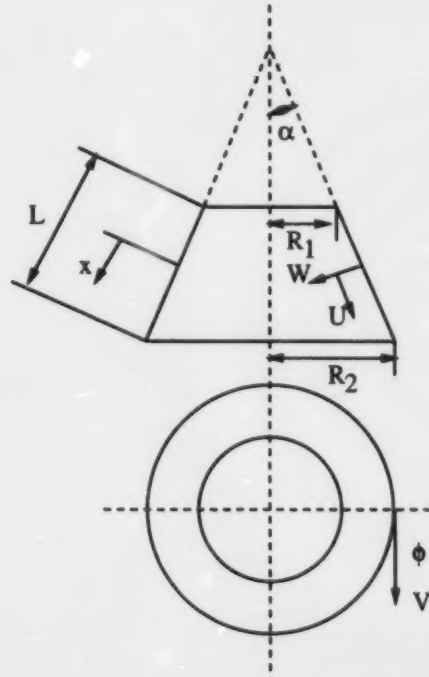


Figure 1: Geometry and notations

ACCURATE FREE VIBRATION ANALYSIS OF CLAMPED MINDLIN PLATES USING THE METHOD OF SUPERPOSITION

S.D. Yu and W.L. Cleghorn

Department of Mechanical Engineering
University of Toronto
Toronto, Ontario,
Canada M5S 1A4

ABSTRACT

The method of superposition is further developed to analyze free flexural vibration of clamped rectangular Mindlin plates. Comparison of results given by Mindlin theory of plates with those previously obtained by Reissner theory has shown that the rotatory inertia does not significantly affect plate flexural vibration. Accurate eigenvalues are presented for a number of values of plate aspect ratio along with two representative values of thickness ratio.

NOMENCLATURE

a, b	plate dimensions
D	flexural rigidity of plate $= Eh^3/12(1-\nu^2)$
h	plate thickness
K	number of terms used in series expansion
M_x, M_{xy}, M_y	bending and twisting moments
$M_\xi, M_{\xi\eta}, M_\eta$	nondimensional bending and twisting moments
Q_x, Q_y	shear forces on plane $x=const.$ and $y=const.$
Q_ξ, Q_η	corresponding nondimensional shear forces
W	lateral displacement divided by side length a
η	dimensionless distance along y axis $(= y/b)$
λ^2	$= \omega a^2 \sqrt{\rho/D}$, eigenvalue
ν	Poisson ratio
ξ	dimensionless distance along x axis $(= x/a)$
ρ	mass of plate per unit area
ϕ	plate aspect ratio $(= b/a)$
ϕ_h	plate thickness ratio $(= h/a)$
ψ_x, ψ_y	rotations with respect to x and y , respectively
ψ_ξ, ψ_η	rotations with respect to ξ and η , respectively
ω	circular frequency of plate vibration

INTRODUCTION

The classical plate theory (CPT), developed by Kirchhoff and Lord Kelvin, is based on the hypothesis that material lines normal to the midplane of a plate before deformation remain straight and normal to the midplane after deformation. Since shear deformation and rotatory inertia are neglected in CPT, eigenvalues for free vibration of plates are acceptable only for thin plates. When the thickness of a plate is significant, effects of shear deformation must be considered. To investigate static behavior of shear-deformable plates, Reissner [1] introduced his theory of plate bending. Mindlin [2] included the effect of rota-

tory inertia in treating plate dynamics, and fulfilled the so-called improved plate theory (IPT). A shear correction factor κ^2 was introduced in his paper to account for the variations of shear stresses across the thickness. Mindlin, Schacknow and Deresiewicz [3] also investigated the free vibration problem of thick rectangular plates with one pair of opposite edges simply supported by using Lévy technique. Lee and Reismann [4] gave an exact solution for the vibration of simply supported rectangular plates. Palardy and Palazotto [5] applied the Lévy method to buckling and vibration of composite plates by using the Mindlin theory of plates.

Reviewing the literature, it is noticed that very little has been done regarding the accurate analytical solutions for free vibration of thick rectangular plates with arbitrary boundary conditions since Mindlin established the improved plate theory. The two exceptions are simply supported rectangular plates, and plates with a pair of edges simply supported while the other two edges remain unrestricted. The Navier method is used to formulate solutions for the simply supported plates. Solutions for plates with a pair of opposite edges simply supported may be obtained by the Lévy method.

Besides the Navier method and the Lévy method, several other approaches have been used to obtain approximate solutions for thick rectangular plates. Among them the Rayleigh-Ritz and the finite element methods are commonly used in the free vibration analysis. Dawe and Roufaeil [6] investigated free vibration of Mindlin plates with several boundary conditions by the Rayleigh-Ritz method. Researchers interested in energy methods are invited to refer to a book by Reddy [7] for detailed description.

In this paper, the superposition method, developed by Gorman [8] for thin plates, is now employed to obtain analytical solutions for rectangular Mindlin plates. It has been shown that this method works extremely well for thin rectangular plates where shear deformation and rotatory inertia are typically neglected. Recently Cleghorn and Yu [9] considered the effects of shear

deformation on the free vibration of rectangular plates by the method of superposition. A comparison with the results in [9] will be further referred to in this paper for understanding of the effects of rotatory inertia on eigenvalues of free flexural vibration.

Since analytical expressions in conjunction with the superposition method are highly dependent on the boundary conditions imposed along the four rectilinear edges, one has to use different building blocks for plates with different boundary conditions. To overcome this disadvantage, Yu and Cleghorn [10] proposed the use of four building blocks and demonstrated that one needs to investigate only six cases of support in order to acquire information on the free vibration of plates with arbitrary classical boundary conditions, where plates with four edges simply supported are excluded. In this paper, a fully clamped thick rectangular plate is studied in detail by the superposition method. Results for plates with a combination of clamped and simply supported edges may be obtained from those for the clamped plates by eliminating contributions of the building blocks associated with simply supported edges.

GOVERNING DIFFERENTIAL EQUATIONS

The governing differential equations for plate free vibration while shear and rotatory inertia are taken into consideration according to Reissmann [11], may be written as,

$$\begin{aligned} \kappa^2 G h \left[\nabla^2 w + \frac{\partial \psi_x}{\partial x} + \frac{\partial \psi_y}{\partial y} \right] + \rho \omega^2 w &= 0 \\ \frac{D}{2} \left[2 \frac{\partial^2 \psi_x}{\partial x^2} + (1-\nu) \frac{\partial^2 \psi_x}{\partial y^2} + (1+\nu) \frac{\partial^2 \psi_y}{\partial x \partial y} \right] \\ - \kappa^2 G h \left[\psi_x + \frac{\partial w}{\partial x} \right] + \frac{\rho \omega^2 h^2}{12} \psi_x &= 0 \\ \frac{D}{2} \left[2 \frac{\partial^2 \psi_y}{\partial y^2} + (1-\nu) \frac{\partial^2 \psi_y}{\partial x^2} + (1+\nu) \frac{\partial^2 \psi_x}{\partial x \partial y} \right] \\ - \kappa^2 G h \left[\psi_y + \frac{\partial w}{\partial y} \right] + \frac{\rho \omega^2 h^2}{12} \psi_y &= 0 \end{aligned} \quad (1)$$

where definitions of variables appearing in the above equations are given in the Nomenclature.

The experience has shown that it is very advantageous to nondimensionalize all mechanical quantities and governing differential equations. Accordingly, we introduce the following transformations

$$\begin{aligned} \xi = \frac{x}{a}, \quad \eta = \frac{y}{b}, \quad W(\xi, \eta) = \frac{w(x, y)}{a} \\ M_\xi = \frac{M_x a}{D}, \quad M_\eta = \frac{M_y b}{D}, \quad M_{\xi\eta} = \frac{2M_{xy} a}{D(1-\nu)} \\ Q_\xi = \frac{Q_x}{\kappa^2 G h}, \quad Q_\eta = \frac{Q_y}{\kappa^2 G h} \end{aligned} \quad (2)$$

The above partial differential equations may be rewritten as

$$\frac{\partial^2 W}{\partial \xi^2} + \frac{1}{\phi^2} \frac{\partial^2 W}{\partial \eta^2} + \frac{\partial \Psi_\xi}{\partial \xi} + \frac{1}{\phi} \frac{\partial \Psi_\eta}{\partial \eta} + \frac{\lambda^4 \phi_h^2}{\nu_3} W = 0$$

$$\frac{\partial^2 \Psi_\xi}{\partial \xi^2} + \frac{\nu_1}{\phi^2} \frac{\partial^2 \Psi_\xi}{\partial \eta^2} + \frac{\nu_2}{\phi} \frac{\partial^2 \Psi_\eta}{\partial \xi \partial \eta} - \frac{\nu_3}{\phi_h^2} \left(\Psi_\xi + \frac{\partial W}{\partial \xi} \right) + \frac{\lambda^4 \phi_h^2}{12} \Psi_\xi = 0$$

$$\frac{\partial^2 \Psi_\eta}{\partial \xi^2} + \frac{1}{\phi^2 \nu_1} \frac{\partial^2 \Psi_\eta}{\partial \eta^2} + \frac{\nu_2}{\phi \nu_1} \frac{\partial^2 \Psi_\xi}{\partial \xi \partial \eta} - \frac{\nu_3}{\phi_h^2 \nu_1} \left(\Psi_\eta + \frac{1}{\phi} \frac{\partial W}{\partial \eta} \right) + \frac{\lambda^4 \phi_h^2}{12 \nu_1} \Psi_\eta = 0 \quad (3)$$

where, $\phi_h = \frac{a}{h}$, $\nu_1 = \frac{1-\nu}{2}$, $\nu_2 = \frac{1+\nu}{2}$, $\nu_3 = 6\kappa^2(1-\nu)$. The corresponding nondimensional shearing forces, bending and twisting moments become

$$\begin{aligned} Q_\xi &= \Psi_\xi + \frac{\partial W}{\partial \xi}, \quad Q_\eta = \Psi_\eta + \frac{1}{\phi} \frac{\partial W}{\partial \eta} \\ M_\xi &= \frac{\partial \Psi_\xi}{\partial \xi} + \frac{\nu}{\phi} \frac{\partial \Psi_\eta}{\partial \eta}, \quad M_\eta = \frac{\partial \Psi_\eta}{\partial \eta} + \nu \phi \frac{\partial \Psi_\xi}{\partial \xi}, \\ M_{\xi\eta} &= \frac{\partial \Psi_\xi}{\partial \eta} + \frac{\partial \Psi_\eta}{\partial \xi} \end{aligned} \quad (4)$$

SOLUTIONS FOR THE EMPLOYED BUILDING BLOCKS

Since the governing differential equations are linear, four building blocks as shown in Fig. 1 may be employed to fulfill the superposition procedure. To employ the superposition technique, we first release the boundary condition $\Psi_n = 0$ along all the four rectilinear edges. Analytical solutions for each building block may be found because there are three simply supported edges in all of the building blocks. Eigenvalues and mode shapes may then be obtained by superimposing solutions for the four building blocks and requiring the superimposed solutions to satisfy the released boundary conditions.

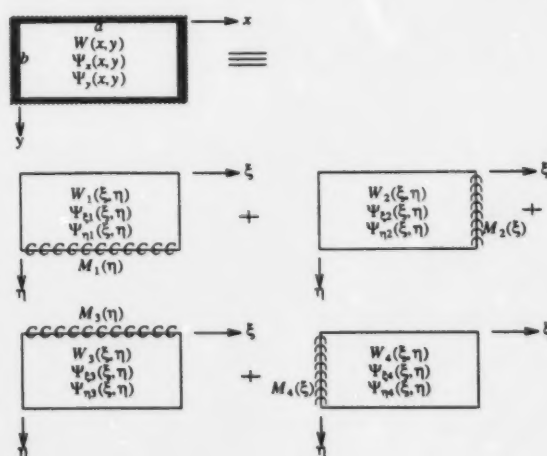


Fig. 2 Building blocks used for vibration analysis of clamped Mindlin plates

Solution for the First Building Block

The first building block has simply supported boundary conditions along the edges $\xi=0$, $\xi=1$ and $\eta=0$. On the edge $\eta=1$, the lateral displacement and rotation angle Ψ_ξ vanish. The amplitude of the driving bending moment is given by

$$\frac{M_1(\xi)b}{D} = \sum_{m=1}^K U_m \sin m\pi\xi \quad (5)$$

Lévy type solutions for the lateral displacement and rotation angles satisfying the imposed boundary conditions along the two opposite edges $\xi=0$ and $\xi=1$ may be written as

$$\begin{aligned} W_1(\xi, \eta) &= \sum_{m=1}^K X_m(\eta) \sin m\pi\xi \\ \Psi_{\xi 1}(\xi, \eta) &= \sum_{m=1}^K Y_m(\eta) \cos m\pi\xi \\ \Psi_{\eta 1}(\xi, \eta) &= \sum_{m=1}^K Z_m(\eta) \sin m\pi\xi \end{aligned} \quad (6)$$

Substituting the above Lévy type solutions into Eqn.(3), we obtain

$$\begin{Bmatrix} X_m'' \\ Y_m'' \\ Z_m'' \end{Bmatrix} + \begin{bmatrix} 0 & 0 & a_{m1} \\ 0 & 0 & a_{m2} \\ a_{m3} & a_{m4} & 0 \end{bmatrix} \begin{Bmatrix} X_m' \\ Y_m' \\ Z_m' \end{Bmatrix} + \begin{bmatrix} b_{m1}b_{m2} & 0 \\ b_{m3}b_{m4} & 0 \\ 0 & 0 & b_{m5} \end{bmatrix} \begin{Bmatrix} X_m \\ Y_m \\ Z_m \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ 0 \end{Bmatrix} \quad (7)$$

where single and double primes represent the first and second derivatives with respect to η , respectively. The characteristic equation which determines solutions of Eqn. (7) is

$$\epsilon_m^6 + p_{m2}\epsilon_m^4 + p_{m1}\epsilon_m^2 + p_{m0} = 0 \quad (8)$$

To find analytical expressions for the roots of the above characteristic equation, let

$$\epsilon_m^2 = \sigma_m, \quad \sigma_m = \bar{\sigma}_m - \frac{p_{m2}}{3} \quad (9)$$

Substitution of Eqn.(9) into Eqn.(8) gives

$$\bar{\sigma}_m^3 + 3q_m\bar{\sigma}_m + 2r_m = 0 \quad (10)$$

where

$$3q_m = p_{m1} - \frac{p_{m2}^2}{3}, \quad 2r_m = p_{m0} + \frac{2p_{m2}^3}{27} - \frac{p_{m1}p_{m2}}{3}$$

Since q_m and r_m are real, Eqn.(10) has either three real roots, or one real root along with two conjugate complex roots, depending on if the discriminant $\Delta = r_m^2 + q_m^3 = 0$ is negative or positive. If we simply exclude the case $\Delta = 0$, then all three real roots are distinct. Based on this consideration, we need investigate only two general cases, $\Delta < 0$ and $\Delta > 0$. However, for Mindlin plates of isotropic materials, it is found that the discriminant corresponding to Eqn. (10) is always negative. This will allow us to consider only one case $\Delta < 0$ in which Eqn. (10) has three real roots. Roots of Eqn. (8) may be written as

$$\epsilon_{m1,2} = \pm\alpha_m, \quad \text{or } \pm i\alpha_m$$

$$\begin{aligned} \epsilon_{m3,4} &= \pm\beta_m, \quad \text{or } \pm i\beta_m \\ \epsilon_{m5,6} &= \pm\gamma_m, \quad \text{or } \pm i\gamma_m \end{aligned} \quad (11)$$

Analytical solutions may then be written as

$$\begin{aligned} X_m &= A_m CS(\alpha_m\eta) + B_m SN(\alpha_m\eta) + C_m CS(\beta_m\eta) \\ &+ D_m SN(\beta_m\eta) + E_m CS(\gamma_m\eta) + F_m SN(\gamma_m\eta) \end{aligned} \quad (12)$$

$$\begin{aligned} Y_m &= R_{m1} [A_m CS(\alpha_m\eta) + B_m SN(\alpha_m\eta)] \\ &+ R_{m2} [C_m CS(\beta_m\eta) + D_m SN(\beta_m\eta)] \\ &+ R_{m3} [E_m CS(\gamma_m\eta) + F_m SN(\gamma_m\eta)] \end{aligned} \quad (13)$$

$$\begin{aligned} Z_m &= S_{m1} [\pm B_m CS(\alpha_m\eta) + A_m SN(\alpha_m\eta)] \\ &+ S_{m2} [\pm D_m CS(\beta_m\eta) + C_m SN(\beta_m\eta)] \\ &+ S_{m3} [\pm F_m CS(\gamma_m\eta) + E_m SN(\gamma_m\eta)] \end{aligned} \quad (14)$$

where, CS represents the cosine, or hyperbolic cosine; and SN represents the sine, or hyperbolic sine, whichever is applicable. Sine and cosine functions apply when characteristic roots of Eqn. (8) are purely imaginary, and hyperbolic sine and hyperbolic cosine functions apply when they are real. In Eqn. (14), plus sign is applicable to hyperbolic cosine and minus sign is applicable to cosine.

Since analytical solutions for the first building block are available, we are now in a position to determine the unknown constants, A_m , B_m , C_m , D_m , E_m and F_m . To achieve this we need to consider the boundary conditions imposed on the first building block. Recalling the description on boundary conditions at the beginning of this section, we are able to derive boundary conditions for the unknown functions $X_m(\eta)$, $Y_m(\eta)$ and $Z_m(\eta)$. They are

$$X_m(0) = X_m(1) = Y_m(0) = Y_m(1) = Z_m'(0) = 0, \quad Z_m'(1) = U_m \quad (15)$$

Considering the above boundary conditions, one is able to determine all six sets of unknown constants appearing in Eqns.(12)-(14). The solutions are

$$W_1 = \sum_{m=1}^K U_m [\theta_{11m} SN(\alpha_m\eta) + \theta_{12m} SN(\beta_m\eta) + \theta_{13m} SN(\gamma_m\eta)] \sin m\pi\xi \quad (16)$$

$$\Psi_{\xi 1} = \sum_{m=1}^K U_m [\theta_{21m} SN(\alpha_m\eta) + \theta_{22m} SN(\beta_m\eta) + \theta_{23m} SN(\gamma_m\eta)] \cos m\pi\xi \quad (17)$$

$$\Psi_{\eta 1} = \sum_{m=1}^K U_m [\theta_{31m} SN(\alpha_m\eta) + \theta_{32m} SN(\beta_m\eta) + \theta_{33m} SN(\gamma_m\eta)] \sin m\pi\xi \quad (18)$$

where R_{mi} , S_{mi} and θ_{ijm} are known constants in terms of coefficients of differential equations and roots of characteristic equations.

Solution for the second building block

The second building block has simply supported boundary conditions along the edges $\eta = 0$, $\eta = 1$ and $\xi = 0$. Along the edge

$\xi = 1$ it has zero lateral displacement, zero rotation angle Ψ_η and a bending moment of the amplitude

$$\frac{aM_2(\eta)}{D} = \sum_{n=1}^K E_n \sin n\pi\eta \quad (19)$$

Lévy type solutions for lateral displacement and two angles of rotation which satisfy the boundary conditions along $\xi = 0$ and $\xi = 1$ may be written as

$$W_2(\xi, \eta) = \sum_{n=1}^K X_n(\xi) \sin n\pi\eta \quad (20)$$

$$\Psi_{\xi 2}(\xi, \eta) = \sum_{n=1}^K Z_n(\xi) \sin n\pi\eta \quad (21)$$

$$\Psi_{\eta 2}(\xi, \eta) = \sum_{n=1}^K Y_n(\xi) \cos n\pi\eta \quad (22)$$

Solutions of $X_n(\xi)$, $Y_n(\xi)$ and $Z_n(\xi)$ are obtained by a procedure similar to that used for the first building block. For completeness, they are written as

$$W_2 = \sum_{n=1}^K U_n \left[\theta_{11n} SN(\alpha_n \eta) + \theta_{12n} SN(\beta_n \eta) + \theta_{13n} SN(\gamma_n \eta) \right] \sin n\pi\xi \quad (23)$$

$$\Psi_{\eta 2} = \sum_{n=1}^K U_n \left[\theta_{21n} SN(\alpha_n \xi) + \theta_{22n} SN(\beta_n \xi) + \theta_{23n} SN(\gamma_n \xi) \right] \cos n\pi\eta \quad (24)$$

$$\Psi_{\xi 2} = \sum_{n=1}^K U_n \left[\theta_{31n} SN(\alpha_n \xi) + \theta_{32n} SN(\beta_n \xi) + \theta_{33n} SN(\gamma_n \xi) \right] \sin n\pi\eta \quad (25)$$

Solution for the third and fourth building blocks

Solutions for the third and fourth building blocks may be inferred from those for the first and second building blocks, respectively. One can simply obtain solutions for the third building block by changing η for $1-\eta$ in solutions for the first building block. All subscripts containing m are subsequently replaced by p . Solutions for the fourth building block may be obtained by changing ξ for $1-\xi$ in solutions for the second building. At this time, subscript n is replaced by q .

DEVELOPMENT OF EIGENVALUE MATRIX

Solutions for the fully clamped plates may be obtained by superimposing solutions for the four building blocks. They are written as

$$W(\xi, \eta) = \sum_{m=1}^K X_m(\eta) \sin m\pi\xi + \sum_{n=1}^K X_n(\xi) \sin n\pi\eta + \sum_{p=1}^K X_p(1-\eta) \sin p\pi\xi + \sum_{q=1}^K X_q(1-\xi) \sin q\pi\eta \quad (26)$$

$$\Psi_\xi(\xi, \eta) = \sum_{m=1}^K Y_m(\eta) \cos m\pi\xi + \sum_{n=1}^K Z_n(\xi) \sin n\pi\eta + \sum_{p=1}^K Y_p(1-\eta) \cos p\pi\xi + \sum_{q=1}^K Z_q(1-\xi) \sin q\pi\eta \quad (27)$$

$$\Psi_\eta(\xi, \eta) = \sum_{m=1}^K Z_m(\eta) \sin m\pi\xi + \sum_{n=1}^K Y_n(\xi) \cos n\pi\eta + \sum_{p=1}^K Z_p(1-\eta) \sin p\pi\xi + \sum_{q=1}^K Y_q(1-\xi) \cos q\pi\eta \quad (28)$$

The above solutions contain four sets of constants, U_m , U_n , U_p and U_q . Each set consists of K constants. These $4K$ unknowns may be determined by requiring the boundary conditions released during the building block constructions to be exactly satisfied. These four conditions are

$$\Psi_\eta(\xi, 1) = \Psi_\xi(1, \eta) = \Psi_\eta(\xi, 0) = \Psi_\xi(0, \eta) = 0 \quad (29)$$

Consideration of these four boundary conditions will result in $4K$ homogeneous algebraic equations relating unknowns E_m , E_n , E_p and E_q . Eigenvalues are then obtained by seeking those λ^2 which cause the determinant of the coefficient matrix to vanish.

PRESENTATION OF NUMERICAL RESULTS

In running the Fortran codes for the entitled vibration analysis, it is found that the rate of convergence for the first six eigenvalues is excellent. Four digits of accuracy for all computed data may be obtained when ten terms in Lévy type solutions are used. Fig. 1 shows the convergence trend for the first two vibration modes of a square plate. It can be seen from Fig. 1 that the eigenvalues converge monotonically to the exact values from the low side. For the first mode, the eigenvalue given by the superposition method converges so fast that the error is even within 1% when only one term is used in the Lévy type expressions.

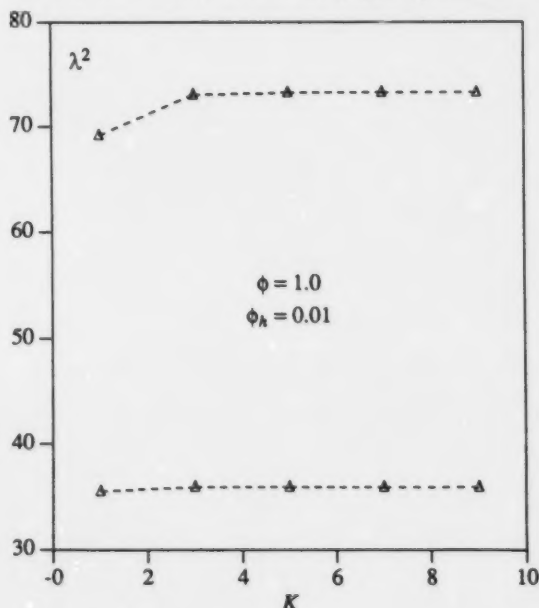


Fig. 2 Convergence trend of the first two vibration modes

To make comparison with the available data in the literature, calculations are performed for the Poisson ratio $\nu=0.3$ and Mindlin's shear factor $\kappa^2=0.8601$. Table 1 presents results for a clamped square plate from references [6] and the present study. It can be seen that results between Dawe and Roufaeil [6] and this paper are in excellent agreement. However, their eigenvalues are slightly higher because the Rayleigh-Ritz energy method is used in their analysis.

Table 2 shows eigenvalues for a square plate from the Mindlin theory, and the Reissner theory by including flexural inertia force. It is fairly obvious that rotatory inertia does not significantly affect eigenvalues of plates. However, consideration of shear effects is necessary for thick plates, especially for those higher vibration modes.

Table 3 gives the first six eigenvalues of clamped rectangular plates with $\phi_h=0.01$ along with a number of representative plate aspect ratios. Table 4 shows the eigenvalues for $\phi_h=0.1$. All data prepared for Tables 1 and 2 possess four digits of accuracy. A quick comparison between these two tables reveals that deviation of results based on classical plate theory from those based on Mindlin plate theory is significant when the plate thickness ratio goes higher.

Table 1 Comparison of eigenvalue for a clamped square plate ($\nu=0.3$, $\kappa^2=0.8601$)

mode	$\phi_h=0.01$		$\phi_h=0.1$	
	Dawe [6]	Present	Dawe [6]	Present
1	35.95	35.94	32.67	32.61
2	73.29	73.23	62.42	62.28
3	73.29	73.23	62.42	62.28
4	108.1	107.9	87.82	87.35
5	131.2	131.1	103.2	103.0
6	131.8	131.8	104.1	103.9

Table 2 Comparison of eigenvalues obtained by Mindlin's and Reissner's theories for a clamped square plate ($\nu=0.3$, $\kappa^2=0.8601$)

mode	$\phi_h=0.01$		$\phi_h=0.1$		Classical
	Reissner	Mindlin	Reissner	Mindlin	
1	35.95	35.94	32.80	32.61	35.99
2	73.26	73.23	63.28	62.28	73.39
3	73.26	73.23	63.28	62.28	73.39
4	108.0	107.9	89.20	87.35	108.2
5	131.2	131.1	106.0	103.0	131.6
6	131.8	131.8	107.0	103.9	132.2

Table 3 Eigenvalues λ^2 for clamped rectangular plates ($\phi_h=0.01$, $\nu=0.3$, $\kappa^2=0.8601$)

mode	plate aspect ratio ϕ					
	1.00	1.25	1.50	2.00	2.50	3.00
1	35.94	29.85	26.97	24.55	23.62	23.17
2	73.23	52.42	41.64	31.79	27.77	25.83
3	73.23	68.37	66.00	44.70	35.37	30.70
4	107.9	89.04	66.40	63.22	46.60	38.04
5	131.1	89.12	79.62	63.86	61.38	47.88
6	131.8	123.9	100.6	70.93	62.96	60.19

Table 4 Eigenvalues λ^2 for clamped rectangular plates ($\phi_h=0.1$, $\nu=0.3$, $\kappa^2=0.8601$)

mode	plate aspect ratio ϕ					
	1.00	1.25	1.50	2.00	2.50	3.00
1	32.61	27.47	24.97	22.84	22.01	21.62
2	62.28	46.35	37.58	29.19	25.68	23.97
3	62.28	58.64	56.83	40.28	32.34	28.27
4	87.35	74.24	57.72	55.18	41.97	34.69
5	103.0	74.62	67.27	55.50	54.20	43.12
6	103.9	98.92	83.49	60.68	57.84	53.35

CONCLUSIONS

This paper presents how the superposition method is employed to perform accurate free vibration analysis of fully clamped rectangular Mindlin plates. It is found that the method presented is applicable to rectangular Mindlin plates with any other classical boundary conditions and even more complicated supports.

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Elastic-Plastic Analysis and Experiment on Expanded Joints of Aluminium Tube to Steel Tubesheets

Y.Z.Zhu

Z.F.Sang

Mechanical Engineering Department
Nanjing Institute of Chemical Technology, Nanjing 210009
P.R.China

Abstract

The joint of aluminium tube to steel tubesheet has long been a key problem in the manufacture of aluminium heat exchangers, to which an immediate solution is needed. In the present work the aluminium tube to steel tubesheet joint was accomplished by lining the inside of the aluminium tube with a thin steel tube and then connecting with steel tubesheet through expanded joints. Theoretical analysis of expanded joint based on elastic-plastic theory and expansion experiment on pull-out strength of this joint show that this expanded joint possesses excellent pull-out strength. Therefore it can meet the demands of the manufacturing industry.

Key words: Elastic-plastic Analysis, Expansion, Pull-out Strength, Aluminium Heat Exchanger.

Nomenclature

$\sigma_{11}, \sigma_{12}, \sigma_{13}$ = yielding limits of liner tube, aluminium tube, and tubesheet material respectively MPa
 E_1, E_2, E_3 = modulus of elasticity of liner tube, aluminium tube, and tubesheet material MPa
 μ_1, μ_2, μ_3 = poisons ratio of liner tube, aluminium tube, and tubesheet material
 r_{11}, r_{10} = internal and external radius of liner tube mm
 r_{21}, r_{20} = internal and external radius of aluminium tube mm
 r_{31}, r_{30} = internal and external radius of tubesheet model mm
 D_{31}, D_{30} = internal diameter of tubesheet model before and after expansion mm
 P_1 = pressure at internal wall of liner tube mm
 P'_{10}, P'_{10} = contact pressure between liner tube and aluminium tube during loading and unloading MPa
 P'_{20}, P'_{20} = contact pressure between aluminium tube and tubesheet hole during loading and unloading MPa
 P_{CG} = residual contact pressure between alumi-

nium tube and tubesheet hole after unloading MPa

P = pull-out force of expanded joints KN

f = friction factor

l = length of expanded joints mm

$\delta'_{r10}, \delta'_{r10}$ = deformation amount at external diameter of liner tube under P_1, P'_{10} respectively mm

$\delta'_{r21}, \delta'_{r21}$ = deformation amount at internal diameter of aluminium tube under P'_{10}, P'_{20} respectively mm

$\delta'_{r20}, \delta'_{r20}$ = deformation amount at external diameter of aluminium tube under P'_{10}, P'_{20} respectively mm

$\delta'_{r31}, \delta'_{r31}$ = deformation amount at internal diameter of tubesheet hole under P'_{20}, P'_{20} respectively mm

Introduction

Aluminium heat exchangers are widely used in natural gas liquefying, air separation, low temperature and resistance corrosion equipment. In order to reduce the thickness of the tubesheet for the heat exchangers, steel tubesheets are commonly used. However steel and aluminium differ much in modulus of

elasticity, strength and thermal conductivity. This makes tube-to-tubesheet joint difficult. If welding is employed, problem remains that aluminium tubes will melt at certain welding current while at the same time steel tubesheets have not reached the melting point. When expanded joint technique is used, aluminium tube having produced plastic flow does not enable steel tubesheet to gain deformation amount needed. Therefore the connection technique of aluminium tube to steel tubesheet becomes a critical but unsettled problem in the production of aluminium heat exchangers.

This paper describes an experimental work on the connection between aluminium tube and steel tubesheet. Before aluminium tube is expanded onto steel tubesheet, a thin-walled steel tube is lined to the inside of aluminium tube shown in Fig.1. An elastic-plastic analysis was done for the expanded joints and experiment was performed using tube roller. Results indicate that the expanded joint between aluminium tube lined thin-walled steel tube and steel tubesheet possesses a excellent expanded strength. The pull-out strength is larger than ultimate strength of aluminium tube under general expanding level.

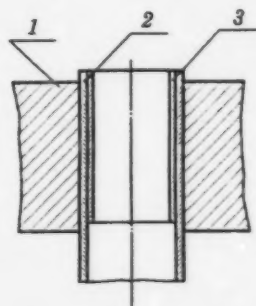


Fig.1 Expanded joint
1.tubesheet, 2. aluminium tube
3.liner tube

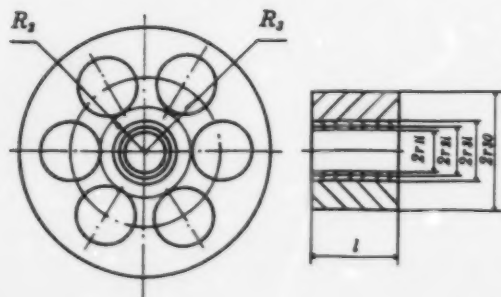


Fig.2 Calculation model

Theoretical Analysis

Expanded joints discussed in this paper is shown in Fig.1. Calculation model used in theoretical analysis is given in

$$\text{Fig.2, } r_{30} = \frac{1}{2} (R_2 + R_3)$$

Formula deduction is based on the following hypotheses :

- (1).Materials for tube and tubesheet are ideal plastic materials in accordance with Von Mises yielding criterion
- (2).Yielding occurs before liner tube contacts aluminium tube, and aluminium tube contacts tubesheet hole (In most cases)
- (3).Discontinuous extruding of mechanical rolling is dealt with as even and continuous.

When liner tube is contacted to aluminium tube, required loading in liner tube is given as

$$P_i = \frac{2}{\sqrt{3}} \sigma_{s1} \ln K_1 \quad (K_1 = \frac{r_{10}}{r_{11}}) \quad (1)$$

Loading further, contact pressure at interface between aluminium tube and tubesheet hole is shown as follows :

$$P_i = P'_{10} + \frac{2}{\sqrt{3}} \sigma_{s1} \ln K_1 \quad (2)$$

$$\text{where } P'_{10} = \frac{2}{\sqrt{3}} \sigma_{s2} \ln K_2 \quad (K_2 = \frac{r_{20}}{r_{21}})$$

If elastic or local plastic deformation occurs around tubesheet hole, internal load of inside tube is defined as :

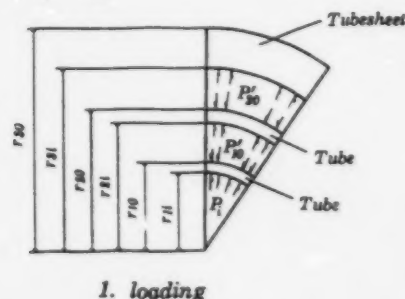
$$P_i = \frac{2}{\sqrt{3}} (\sigma_{s1} \ln K_1 + \sigma_{s2} \ln K_2) + P'_{20} \quad (3)$$

when elastic deformation occurs around tubesheet hole only. In equation (3) :

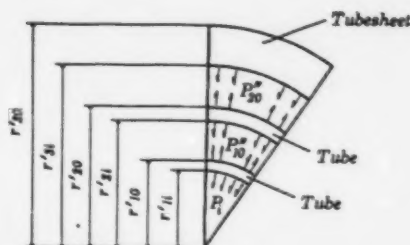
$$P'_{20} = \frac{E_3 \delta \frac{P'_{20}}{r_{21}}}{r_{21} \left(\frac{r_{30}^2 + r_{21}^2}{r_{30}^2 - r_{21}^2} + \mu_3 \right)} \quad (4)$$

Where P'_{20} is the interface pressure between tube and tubesheet hole during loading. However what is of concern is the residual interface pressure (radial compressive stress) between tube and tubesheet hole after unloading. According to the theory of elasticity

$$P_{CG} = P'_{20} - P_{20} \quad (5)$$



1. loading



2. unloading

Fig.3 Pressure on tube and tubesheet during loading and unloading

With a view to the fact that liner tube and aluminium tube, aluminium tube and tubesheet hole are always connected with each other, whether after loading or unloading, compatible equation for deformation between tube and tubesheet hole is expressed as

$$\delta_{r21}^{P_{20}} = \delta_{r20}^{P_{20}} - \delta_{r10}^{P_{20}} \quad (6)$$

$$\delta_{r21}^{P_{10}} - \delta_{r20}^{P_{10}} = \delta_{r10}^{P_{10}} - \delta_{r11}^{P_{10}} \quad (7)$$

where:

$$\delta_{r21}^{P_{20}} = \frac{P_{20} r_{21}}{E_2} \left(\frac{r_{20}^2 + r_{21}^2}{r_{20}^2 - r_{21}^2} + \mu_2 \right) \quad (8)$$

$$\delta_{r20}^{P_{20}} = \frac{P_{20} r_{20}}{E_2} \left(\frac{r_{20}^2 + r_{21}^2}{r_{20}^2 - r_{21}^2} - \mu_2 \right) \quad (9)$$

$$\delta_{r10}^{P_{10}} = \frac{2P_{10} r_{10}^2 r_{20}}{E_1 (r_{20}^2 - r_{21}^2)} \quad (10)$$

$$\delta_{r21}^{P_{10}} = \frac{P_{10} r_{21}}{E_2} \left(\frac{r_{20}^2 + r_{21}^2}{r_{20}^2 - r_{21}^2} + \mu_2 \right) \quad (11)$$

$$\delta_{r11}^{P_{10}} = \frac{P_{10} r_{11}}{E_1} \left(\frac{r_{10}^2 + r_{11}^2}{r_{10}^2 - r_{11}^2} - \mu_1 \right) \quad (12)$$

$$\delta_{r20}^{P_{10}} = -\frac{2P_{10} r_{20}^2 r_{21}}{E_2 (r_{20}^2 - r_{21}^2)} \quad (13)$$

$$\delta_{r10}^{P_{10}} = \frac{2P_{10} r_{11}^2 r_{10}}{E_1 (r_{10}^2 - r_{11}^2)} \quad (14)$$

$$r_{10} \approx r_{21}, \quad r_{20} \approx r_{31}$$

Substituting equation (8), (9), (10) into equation (6), equation (11), (12), (13), (14) into equation (7), the result is:

$$P_{20} = AP_1 \quad (15)$$

$$P_{10} = BP_{20} = A \cdot B \cdot P_1 \quad (16)$$

Coefficient A and B are related to the geometrical

parameters of the tubes and tubesheets:

$$A = \frac{4r_{11}^2 r_{21}^2}{(r_{20}^2 - r_{21}^2)(r_{20}^2 - r_{11}^2)} \cdot \frac{4r_{20}^2 r_{21}^2}{CD + \frac{E_2}{E_1} (r_{20}^2 - r_{21}^2)^2}$$

$$B = \frac{\frac{E_2}{E_3} \left(\frac{r_{20}^2 + r_{21}^2}{r_{20}^2 - r_{21}^2} + \mu_3 \right) + \left(\frac{r_{20}^2 + r_{21}^2}{r_{20}^2 - r_{21}^2} - \mu_2 \right)}{\frac{2r_{21}^2}{r_{20}^2 - r_{21}^2}}$$

$$C = \frac{E_2}{E_3} \left(\frac{r_{20}^2 + r_{21}^2}{r_{20}^2 - r_{21}^2} + \mu_3 \right) + \left(\frac{r_{20}^2 + r_{21}^2}{r_{20}^2 - r_{21}^2} - \mu_2 \right)$$

$$D = \frac{E_1}{E_2} \left(\frac{r_{20}^2 + r_{21}^2}{r_{20}^2 - r_{21}^2} + \mu_2 \right) + \left(\frac{r_{20}^2 + r_{21}^2}{r_{20}^2 - r_{21}^2} - \mu_1 \right)$$

Residual contact pressure between aluminium tube and tubesheet hole is:

$$P_{CG} = P_{20}' - P_{20} = (1 - A) \frac{E_2 \delta_{r21}^{P_{20}'}}{r_{21} \left(\frac{r_{20}^2 + r_{21}^2}{r_{20}^2 - r_{21}^2} + \mu_3 \right)} - \frac{2}{\sqrt{3}} A (\sigma_{s1} \ln K_1 + \sigma_{s2} \ln K_2) \quad (17)$$

Where all the geometrical measurement refer to those after expansion. $\delta_{r21}^{P_{20}'}$ is related to tube rolling rate.

Pull-out strength for expanded joints can be calculated as

$$P = P_{CG} \cdot \pi r_{21} l \quad (18)$$

Although equation (17) is complicated, a table can be made representing the relation between expansion and pull-out strength of connecting tube to tubesheet of different dimensions and materials, which is valuable in engineering practice. The results obtained through analysis are more suitable in engineering application than those through finite element method.

Test

The experiment for the expanded joint connecting aluminum tube and steel tubesheets, liner tube and steel tubesheets was performed by using roller expander, and the pull-out strength of expanded joint was tested at both normal temperature and elevated temperature about 150°C.

Dimensions and mechanical properties of materials for tube and tubesheet, are shown in Table 1.

Test results for some specimens are shown in the Table 2, where K_1 , K_2 refers to the expansion for the increasing rate of internal diameter of tube and tubesheet hole.

Table 1. Dimensions and mechanical properties of materials for tube and tubesheet

	T u b e		Tubesheet
	Carbon steel	Aluminium	
Nominal O. D. mm	20.8	25	100
Nominal Thickness mm	0.5	2, 2.5	40
Nominal Hole Diameter mm			25.4
Hole Pitch mm			32
Yield Limit MPa	250	50	386.6
Tensile Strength MPa	382	100	520
Modulus of Elasticity MPa	2.02×10^5	0.72×10^5	2.12×10^5
Poisson's Ratio	0.3	0.33	0.3

Table 2. Test results

specimen No.	Tubesheet Hole		Carbon steel Tube		Aluminium Tube		L mm
	D _i mm	D _i mm	2r _{1i} mm	2r _{1i} mm	2r _{2i} mm	2r _{2i} mm	
1-1	25.52	25.521	19.90	20.73	21.00		30.6
1-2	25.00	25.003	20.00	21.06	21.00		28.4
1-3	25.44	25.442	20.00	21.09	21.00		27.0
1-4	25.45	25.452	20.00	21.00	21.00		32.9
1-5	25.40	25.401	20.00	21.00	21.00		30.0
1-6	25.40	25.401	19.95	21.01	21.02		22.0
1-7	25.50	25.505	19.90	21.10	21.00		25.0
1-8	25.45	25.456	19.90	21.10	21.00		23.0
2-1	25.44				20.00	20.70	25.4
2-2	25.46				20.08	20.80	30.0
2-3	25.46				20.00	20.50	22.0
2-4	25.50				20.00	20.90	25.0
3-1	25.42	25.422	19.90	20.80	21.00		25.0
3-2	25.42	25.423	19.90	20.85	21.00		26.0
3-3	25.43	25.433	19.95	20.90	21.02		29.0
3-4	25.42	25.425	19.90	20.90	21.00		26.5

specimen No.	Tube Rolling Rate		Theoretical pull-out strength KN	Test pull-out strength KN	Pull-out strength 150°C KN	
	K ₁ %	K ₂ %				
1-1	0.431	0.04	6.7	5.5		
1-2	3.44	0.11	17	13.4 *		
1-3	1.77	0.08	13.8	12.8 *		
1-4	1.37	0.063	9.4	12.2 *		
1-5	1.57	0.04	7.3	8.2		
1-6	1.73	0.04	4.9	4.0		
1-7	1.96	0.196	28	13.4 *		
1-8	2.16	0.24	35.9	12.0 *		
2-1	1.02			9.5		
2-2	1.02			8.4		
2-3	0.157			4.2		
2-4	1.56			8.0		
3-1	1.1	0.08	12.2			9.3
3-2	1.3	0.118	17.3			11 *
3-3	1.65	0.117	20.0			10 *
3-4	1.49	0.197	30.4			10.8 *

* Tube fracture

$$K_1 = \frac{2 (r_{11}' - r_{11}) - (D_1 - 2r_{20}) - 2 (r_{21} - r_{10})}{D_1} \times 100\%$$

$$K_2 = \frac{D_1' - D_1}{D_1} \times 100\%$$

Summary and Discussion

The results of the experimental study presented here can be summarized as following :

(1) The expansion joint connecting thin steel tube-lined aluminium tube with steel tubesheet exhibits good connection strength, and the pull-out strength of this joint under medium expansion level exceeds that of the aluminium tube alone, while expanded joint between steel tubesheet and aluminium tube without liner tube can hardly achieve this aim.

This technique provides a way to solve the problem as to the expanded joint of tube to tubesheets in which materials for both are quite different in yielding strength.

(2) As indicated in table 2, fracture of tubes first occurs before the theoretical pull-out strength is reached for the most specimens. The reason caused it is that the expanded strength between tube and tubesheet hole is larger than that of tube itself.

(3) By lining the inside of the aluminium tube with a thin-walled steel tube the expanded joint possesses good performance at high temperature.

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LIMIT LOADS OF FRAMED STRUCTURES AND ARCHES USING THE GLOSS R-NODE METHOD

C. P. D. Fernando and R. Seshadri
Industrial Systems Engineering
University of Regina, Regina, Canada.

ABSTRACT

The GLOSS R-Node method of plastic collapse load determination is extended to structural configurations of frames and arches. In this method the structure is discretized into a large number of slices. The r-node stress for each of these slices is determined and the plastic collapse load estimates are then obtained. Comparison of these collapse load estimates with analytical results is encouraging.

NOMENCLATURE

A_1, A_2	representative bar areas in the r-node bar system
A	radius of curvature for an arch
B, n	second stage creep parameters
E_0	modulus of elasticity
E_1, E_2	modulus of elasticity of the bars 1 and 2 in the r-node bar system
E_s	secant modulus
F	proportional constant for applied concentrated load
F_L	limit value of proportional constant
h	wall thickness of cylinder
L_1, L_2	length of bars 1 and 2 in the r-node bar system
L	length of columns/beams
$\bar{M}$	applied bending moment
M_p	plastic bending moment
N	minimum number of hinges required for failure
N_p	collapse load in tension
$\bar{p}$	applied internal pressure
P	applied load
$\bar{P}$	applied axial force
P_L	limit load
$\bar{q}$	applied UDL
$\bar{q}_{collapse}$	analytical collapse load intensity
R_i, R_o	internal/external radii of thick cylinder
R_m	mean radius of the thick-walled cylinder
t	time
x	distance from the fixed end of the beam
β_k	scaling factor at the plastic hinge location k
δ	deflection at a given point in a component or structure
$\gamma, \gamma_1, \gamma_2$	scaling factors
θ	follow-up angle (Fig.1)
ϵ	strain
$\dot{\epsilon}$	strain rate
ϕ_0	semi-included angle of an arch

σ	stress
σ_e	effective or von Mises stress
σ_{ei}	pseudoelastic effective stress in an element
$(\sigma_e)_{r-node}$	r-node effective stress
σ_{nj}	r-node effective stress at the plastic hinge location j
$\bar{\sigma}_n$	combined r-node effective stress
σ_y	yield stress
σ_{ref}	reference stress
ξ	ratio $\frac{\bar{\sigma}_n}{\sigma_y}$
τ_0	specified time
χ	angle measured from horizontal where hinges form in an arch

Subscripts

c	creep
m	mean
p	plastic

INTRODUCTION

The use of limit analysis in the design of components and structures is well documented in the literature [1,2,3]. Designers have come to recognize that structures can withstand loads beyond the elastic limit of structural materials and with plastic design, advantage can be taken of the reserve strength that exists beyond the initial yielding. This reserve strength is significant for statically indeterminate structures, especially those with larger redundancies. Therefore, the knowledge of limit loads of components and structures becomes useful to a designer, since it enables the determination of the reserve strength and addresses the mode of failure associated with load-controlled effects.

Plastic analysis not only produces economies in materials but also simplifies the design procedure through the use of the so-called "bounding theorems". Lack of fit in structural connections, residual stresses and other fabrication defects which are difficult to quantify do not affect plastic analysis whereas it is necessary to take these factors into account when elastic analysis is carried out making it more cumbersome. It is therefore more useful to analyze structures that are on the "verge of collapse" and to then establish appropriate working levels of applied loads.

Limit analysis is not always simple to apply. An examination of the literature on plastic analysis reveals that analytical solutions are available only for simple cases of loadings and geometric

configurations. When analytical solutions are not possible, the practice has been to make use of the upper and lower bound theorems of limit analysis. The lower bound theorem states that "if an equilibrium distribution of stresses can be found which balances the applied load and is everywhere below yield, the structure will not collapse". The theorem in essence states that the structure will withstand the applied load by rearranging the internal stresses to the best advantage, if it is possible. It gives lower bounds on, or safe values of, the limit or collapse loading and the maximum lower bound would be the limit load. The methods which use the lower bound theorem to obtain the limit loads are termed "statical methods".

The upper bound theorem on the other hand can be stated as follows: "The structure must collapse if there is any compatible pattern of plastic deformation for which the rate at which the external forces do work is equal to or exceeds the rate of internal dissipation". The upper bound theorem says that if a path of failure exists, the structure will take the path. It gives upper bounds on the limit load, the minimum upper bound being the limit load. The methods which use the upper bound theorem to obtain the limit loads are termed kinematic methods.

The foregoing methods when applied in conjunction with the associated theorems provide bounds on the limit loads and the values of these depend on the deformation pattern and location of hinge formation assumed. That is, they do not predict the exact locations where hinges form in the structure. However, systematic procedures based on the bounding theorems to obtain limit loads have been devised to obtain reasonable results, although some intuition and experience is required.

The availability of powerful mainframe computers along with nonlinear finite element software packages, has resulted in extensive use of the finite element method. The GLOSS R-Node method [4] is such a method that enables determination of the plastic collapse loads on the basis of two linear elastic finite element analyses.

THEORETICAL BACKGROUND

The Concept of Redistribution Nodes (R-Nodes)

The r-node concept is fundamentally related to the notion of "reference stress" [5] and the ASME stress-classification framework [6]. In this section their interrelationship is discussed, and the methodology for determining plastic collapse loads of mechanical components and structures is discussed.

Soderberg [7] was the first to relate multiaxial component behavior to the behavior of a uniaxial data or a single tensile test. From steady state analysis of creep in thick tubes, an expression for reference stress (σ_{ref}) was obtained as:

$$\sigma_{ref} = \frac{\sqrt{3}}{2} \frac{R_m}{h} \bar{p} \quad (1)$$

where R_m is the mean radius of the tube and h is the wall thickness. Further, the mean radius creep strain is given by

$$\epsilon_{c,m} = \frac{\sqrt{3}}{2} \epsilon_c(\sigma_{ref}) \quad (2)$$

where $\epsilon_c(\sigma_{ref})$ is the uniaxial creep strain of a tensile specimen at the reference stress. Soderberg's work paved the way for the subsequent development of the "reference stress method (RSM)" [5], which essentially attempts to correlate the creep deformations in a structure with the results of an equivalent simple creep test. That is to say, the deflection δ at a point in a structure at some time t_0 is given by:

$$\delta(t_0) = \gamma \epsilon_c(t_0) \quad (3)$$

γ is a geometric scaling factor that depends on the configuration of the structure and the boundary conditions, and $\epsilon_c(t_0)$ is the creep strain at time t_0 as obtained by a creep test performed at a stress level σ_{ref} . It has been observed that the reference stress is insensitive to the exact creep strain rate to stress relationship.

Noting that the reference stress is independent of the creep

exponent, Sim [8] reasoned that as the creep exponent in the relationship $\dot{\epsilon}_c = B\sigma^n$ approaches infinity, the limit solution to perfect plasticity would be approached, i.e.,

$$\sigma_{ref} = \left(\frac{P}{P_L} \right) \sigma_y \quad (4)$$

P is the load on the structure, and P_L is the limit load. The success of Sim's method is based on the availability of limit loads for the configuration under investigation, the determination of which is by no means an easy task for complex geometries and loads.

The concept of r-nodes is useful in the determination of the plastic collapse loads of mechanical components and structures. Furthermore, the concepts of reference stress, r-nodes and limit analysis are related to the fundamental notions of stress-classification. There is an explicit recognition of load and deformation controlled effects in the ASME codes [6]. Load-controlled stresses are statically determinate in that they are induced in order to preserve equilibrium with externally applied forces and moments. Deformation controlled stresses on the other hand are induced as a result of statically indeterminate actions. When inelastic action (plasticity or creep) occurs, the statically indeterminate stresses redistribute throughout the component except at the r-nodes which are essentially locations that are statically determinate. On the GLOSS diagram, Fig. 1, the follow-up angle (θ) would be equal to 90 degrees at the r-nodes [9].

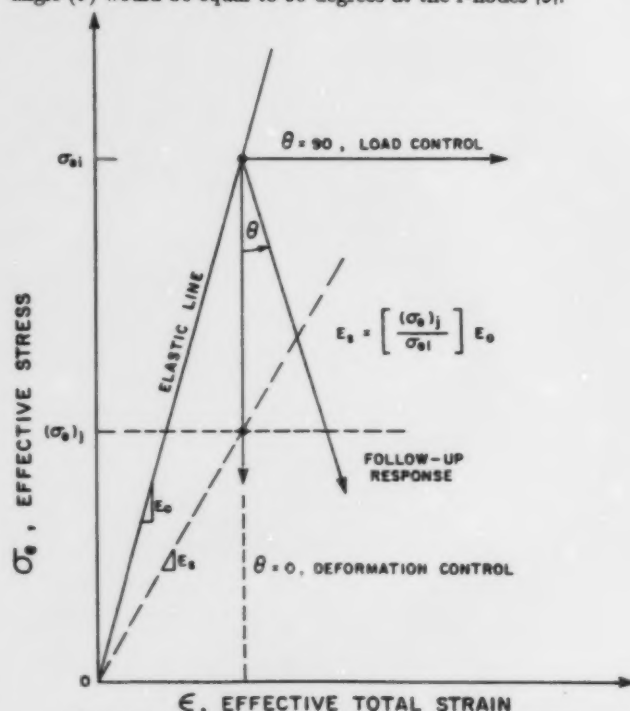


Figure 1: Follow-up Angle (θ) on the GLOSS Diagram

Plastic Collapse of Components and Structures [4]

Since the r-nodes are load-controlled locations within a component or structure, the induced effective stresses are linearly proportional to the externally applied loads or load combinations as a consequence of equilibrium requirements, i.e.,

$$\left. \begin{aligned} (\sigma_e)_{r-nodes} &= \gamma_1 P \\ (\sigma_e)_{r-nodes} &= \gamma_2 < \bar{P}, \bar{M} > \end{aligned} \right\} \quad (5)$$

γ_1 and γ_2 are scaling parameters that would depend on the loading, geometric configuration and material behaviour. For

an elastic perfectly-plastic material, when $(\sigma_e)_{r-node}$ approaches yield stress corresponding to the von Mises criterion, the externally applied load will correspond to the limit load. Equation (5) can therefore be expressed as

$$\left. \begin{aligned} \sigma_v &= \gamma_1 P_L \\ \sigma_v &= \gamma_2 < \tilde{P}, \tilde{M} >_L \end{aligned} \right\} \quad (6)$$

Combining equations (5) and (6),

$$\left. \begin{aligned} P_L &= \left[\frac{\sigma_v}{(\sigma_e)_{r-node}} \right] P \\ < \tilde{P}, \tilde{M} >_L &= \left[\frac{\sigma_v}{(\sigma_e)_{r-node}} \right] < \tilde{P}, \tilde{M} > \end{aligned} \right\} \quad (7)$$

The r-node effective stress, $(\sigma_e)_{r-node}$, can be determined from equilibrium considerations and is therefore insensitive to the inelastic constitutive relationship of the isothermal mechanical component or the structure. The occurrence of a single r-node across the thickness of a component is indicative of a load-controlled membrane mode of collapse. This situation can be represented by a one bar model (Fig. 2) so that plastic collapse would occur when $(\sigma_e)_{r-node} = \sigma_v$ corresponding to the von Mises yield criterion and the formation of a plastic hinge. The presence of a pair of r-nodes is indicative of a load-controlled membrane plus bending mode of plastic collapse. For an elastic-perfectly plastic material, the r-node stresses $(\sigma_e)_{r-node}$ at a given cross-section across the component thickness for a r-node pair are equal in magnitude. Here again, when $(\sigma_e)_{r-node}$ equals yield stress, a plastic hinge will form and the single bar model will apply.

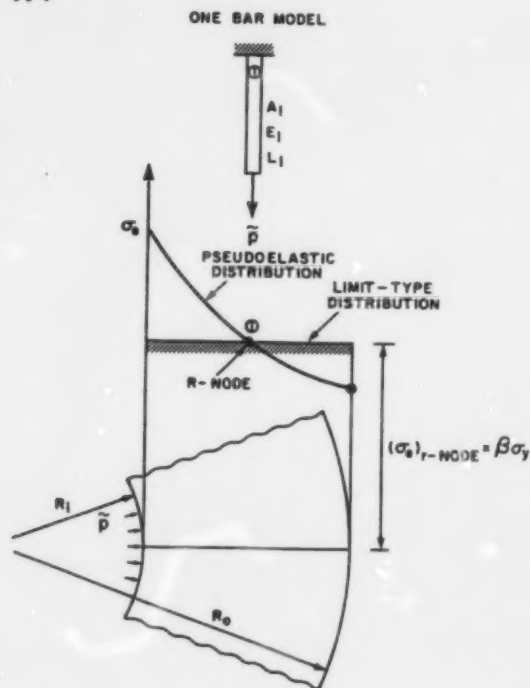


Figure 2 : R-Node Bar Model

For a statically determinate component, the formation of a single plastic hinge will result in its collapse. A typical statically indeterminate mechanical component or structure releases static indeterminacies through the sequential formation of plastic hinges or plastic hinge contours eventually resulting in a collapse mechanism. The number of plastic hinge locations that

would form depends on the degree of static indeterminacy. For example, if a plastic collapse mechanism corresponding to two hinges develops for the beam structure (Fig. 3), then there would be a pair of r-nodes at the hinge locations since bending action is dominant. If σ_{n1} and σ_{n2} are the pseudoelastic effective stresses at the r-nodes corresponding to the plastic hinge locations, then these two r-nodes can be represented by the two bar model. Since r-node stresses are statically determinate, it is sufficient to satisfy external equilibrium, i.e.,

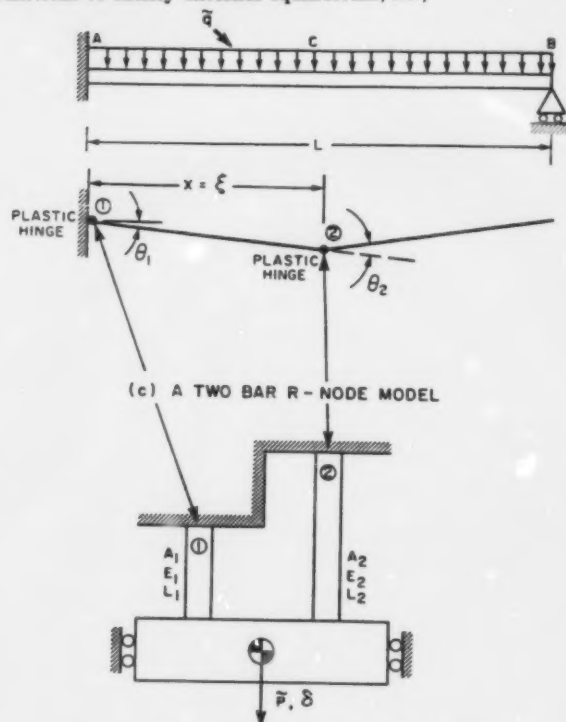


Figure 3 : Collapse Mechanism for a Beam Fixed and Simply Supported with UDL

$$\left. \begin{aligned} \sigma_{n1} A_1 + \sigma_{n2} A_2 &= P \\ \text{or } \sigma_{n1} A_1 + \sigma_{n2} A_2 &= < \tilde{P}, \tilde{M} > \end{aligned} \right\} \quad (8)$$

where A_1 and A_2 are the respective representative areas of the bars corresponding r-nodes 1 and 2.

For a stress distribution corresponding to the plastic collapse mode, the equilibrium requirement can be expressed as

$$\left. \begin{aligned} \bar{\sigma}_n (A_1 + A_2) &= P \\ \text{or } \bar{\sigma}_n (A_1 + A_2) &= < \tilde{P}, \tilde{M} > \end{aligned} \right\} \quad (9)$$

where $\bar{\sigma}_n$ is the "combined r-node effective stress".

From equations (8) and (9),

$$\bar{\sigma}_n = \mu_1 \sigma_{n1} + \mu_2 \sigma_{n2} \quad (10)$$

where $\mu_1 = A_1 / (A_1 + A_2)$ and $\mu_2 = A_2 / (A_1 + A_2)$.

For the two bar mechanism, the pseudoelastic stresses σ_{n1} and σ_{n2} can be obtained as

$$\left. \begin{aligned} \sigma_{n1} &= \frac{E_1/L_1}{(E_1 A_1/L_1) + (E_2 A_2/L_2)} P \\ \sigma_{n2} &= \frac{E_2/L_2}{(E_1 A_1/L_1) + (E_2 A_2/L_2)} P \end{aligned} \right\} \quad (11)$$

In the above equation, the applied load P can be replaced by the applied load combination $< \tilde{P}, \tilde{M} >$ whenever appropriate.

For statically indeterminate structures, the plastic collapse or limit load can be expressed as

$$\left. \begin{aligned} P_L &= \left[\frac{\sigma_2}{\sigma_n} \right] P \\ \text{or } < \tilde{P}, \tilde{M} >_L &= \left[\frac{\sigma_2}{\sigma_n} \right] < \tilde{P}, \tilde{M} > \end{aligned} \right\} \quad (12)$$

Substituting equations (10) and (11) into equation (12), the limit load can be expressed as

$$P_L = \left[\frac{(E_1 A_1 / L_1) + (E_2 A_2 / L_2)}{\mu_1 (E_1 / L_1) + \mu_2 (E_2 / L_2)} \right] \sigma_y \quad (13)$$

As before, P_L in the above equation can be replaced by $< \tilde{P}, \tilde{M} >$ for a load-moment combination. Since the plastic collapse process is essentially load-controlled, it would be independent of the statically indeterminate parameters, i.e., E_1 , E_2 , L_1 and L_2 . Two cases are considered next:

Case 1: $E_1 / L_1 = E_2 / L_2$

Here $P_L = (A_1 + A_2) \sigma_y$ provided $\mu_1 + \mu_2 = 1$. From equations (11)

$$\frac{\sigma_{n1}}{\sigma_{n2}} = \frac{E_1 L_2}{E_2 L_1} = 1 \quad (14)$$

i.e., $\sigma_{n1} = \sigma_{n2}$. This case would have to be regarded as trivial since the two bar model would fail to generate unequal pseudoelastic r-node stresses.

Case 2: $A_1 = A_2 = A_0$ (say)

For this case, $P_L = 2A_0 \sigma_y$ provided $\mu_1 = \mu_2 = 1/2$. The appropriate two bar model is that for which $E_1 = E_2$ and $L_1 \neq L_2$. The combined r-node effective stress or the reference stress can be expressed as the arithmetic average of the pseudoelastic r-node stresses, i.e.,

$$\bar{\sigma}_n = \frac{\sigma_{n1} + \sigma_{n2}}{2} \quad (15)$$

Similarly, for a plastic collapse mechanism involving three plastic hinges,

$$\bar{\sigma}_n = \frac{\sigma_{n1} + \sigma_{n2} + \sigma_{n3}}{3} \quad (16)$$

and the limit load can be obtained using equation (12).

Generalizing the foregoing expressions for N plastic hinges,

$$\bar{\sigma}_n = \frac{\sum_{j=1}^N \sigma_{nj}}{N} \quad (17)$$

N would be one for statically determinate structures, and greater than one for statically indeterminate structures.

Since $\bar{\sigma}_n$ is used in conjunction with equation (12), it would be the reference stress for the given isothermal component or structure. Also $\bar{\sigma}_n$ is statically determinate since it is obtained from a limit-type of stress distribution, and is therefore insensitive to the material parameters. The use of the GLOSS R-Node method for determining reference stresses should open up a wide range of application possibilities [6,10].

DETERMINATION OF PLASTIC HINGE LOCATIONS AND COLLAPSE LOAD OF STRUCTURES.

The GLOSS R-node method can be used to locate the r-nodes in the following manner:

1. A linear elastic finite element analysis of a given mechanical component or structure is carried out for the prescribed isothermal loadings. The resulting stresses would be pseudoelastic quantities.
2. A location j is arbitrarily chosen within the component. The elastic moduli of all other elements are modified according to the equation

$$E_i = \left[\frac{(\sigma_e)_j}{\sigma_{ei}} \right] E_0 \quad (18)$$

Although the choice of this location is somewhat arbitrary, the r-nodes can be located with a reasonable degree of accuracy. The only stipulation for choosing point j is that $(\sigma_e)_j$ should be non-zero. A second linear finite element analysis is then carried out.

3. On the basis of the two linear elastic analysis, the follow up angle θ can be determined for each element. The elements for which $\theta = 90$ degrees can then be identified as the r-node elements. The exact r-node location can be obtained by interpolation.

Alternately, the change in σ_e can be computed for each element, and the elements for which $\Delta \sigma_e = 0$ contain the r-nodes. In practice the foregoing computations can be incorporated into the linear elastic finite element code.

4. The structure can be visualized to be made of a finite number of slices normal to the neutral axis. Since each of these slices is a potential hinge location, each contains a r-node and an associated r-node stress. A plot of these r-node stresses shows peak values at some locations along the structure. The numerical value of these peaks point to the sequence of formation of hinges along the structure and their respective locations. Further, the r-node stresses of the adjacent slices is indicative of the spread of plasticity along the length with increasing external loading.

5. By using the N r-node peak stresses in equation (17), where N is the number of hinges required to form a mechanism in the structure, the average r-node stress for the structure can be obtained. Substituting $\bar{\sigma}_n$ in

$$P_L = \left[\frac{\sigma_y}{\bar{\sigma}_n} \right] P \quad (19)$$

gives the limit load P_L of the structure.

NUMERICAL EXAMPLES

Indeterminate Beam Structures

The indeterminate beam structure shown in Fig. 3 has the end A built in, end B simply supported and a distributed load of intensity $\bar{q}$ applied to it. The first plastic hinge forms at end A and the structure becomes determinate. As the load intensity is increased further, an additional plastic hinge would form at point C and the beam would collapse.

The r-nodes at each plastic hinge location can be represented by uniaxial bars which for determinate structures are coupled as shown in Fig. 3. The coupling of the bars enables the transfer of load to C once the bar A reaches yield. The r-node stress at a plastic hinge location can be expressed as

$$\sigma_{nk} = \beta_k \sigma_y \quad (20)$$

$$(k = 1, 2)$$

β_k is the scaling factor that depends on the geometric configuration and applied loadings. When β_k approaches unity, a plastic hinge would develop at the specific cross-section. The combined r-node effective stress $\bar{\sigma}_n$, which would also be the reference stress for isothermal components and structures can be expressed as

$$\bar{\sigma}_n = \frac{\sigma_{n1} + \sigma_{n2}}{2} \quad (21)$$

where σ_{n1} and σ_{n2} are the pseudoelastic r-node stresses at the plastic hinge locations A and C , respectively. The plastic collapse load intensity can be expressed as

$$\bar{q}_{collapse} = \left(\frac{\sigma_c}{\sigma_n} \right) \bar{q} \quad (22)$$

An expression for the collapse load has been derived analytically by Mendelson [11] as

$$\bar{q}_{collapse} = \frac{11.65685 M_p}{L^2} \quad (23)$$

The point *C* is located at a distance of $\xi = 0.58579$ from the end *A*.

The GLOSS R-Node method can be applied to this indeterminate beam configuration. Here the beam is divided into 100 slices lengthwise and each of these slices is subdivided to give 10 elements per slice across the height of the beam. The plot of r-node stresses for a beam of length 20 inches, cross-section 1 inch $\times$ 1 inch, with an uniformly distributed load of 20 psi, is shown in Fig. 4. Since the degree of indeterminacy is one and two hinges are required to cause collapse, two r-node peak values are substituted in Eq. 21, and the combined r-node effective stress is evaluated. And Eq. 22 then gives the collapse load. Table 1 summarizes the results.

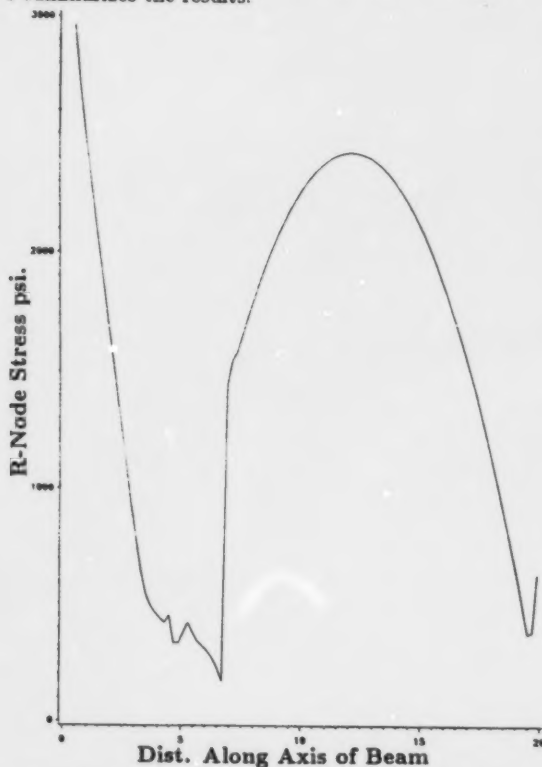


Figure 4 : R-Node Stress Plot for a Beam Fixed and Simply Supported with UDL

A beam with uniformly distributed load fixed at both ends would require the formation of three hinges two at the ends and one at the center (the loading and boundary conditions being symmetrical about the center). The analytical solution is given as [4]:

$$\bar{q}_{collapse} = \frac{16 M_p}{L^2} \quad (24)$$

A plot of the r-node stresses for a beam having identical dimensions and loading as in the previous example, but fixed at both ends is shown in Fig. 5. The peaks are found to occur at either ends and at the center. Using these three peak values, equations

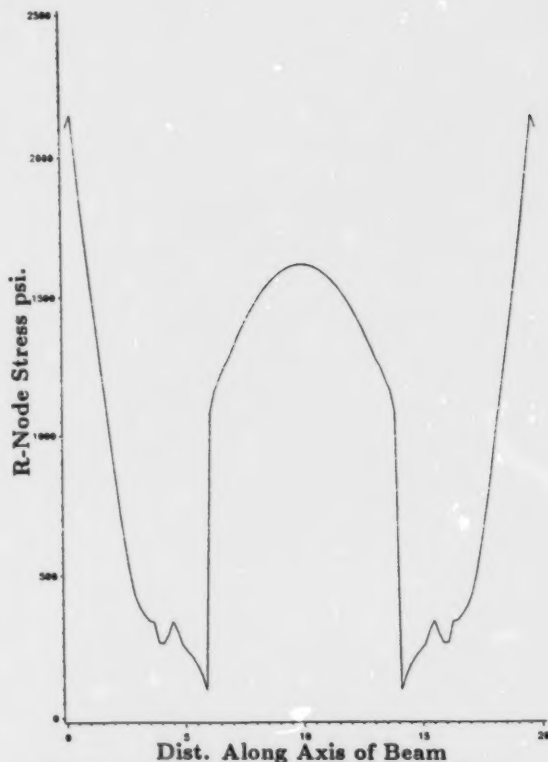


Figure 5 : R-Node Stress Plot for a Beam Fixed at Both Ends with UDL

(17) and (19), the limit load can be determined. The results (Table 1) show that the GLOSS R-Node method predicts the limit loads accurately.

Single and Two Bay Frame Structures

A single bay frame structure with loads applied is shown in Figure 6a. The columns and beam are 10 inches long ($2L$) and the cross-sections are 1 inch $\times$ 1 inch. The various local mechanisms which can cause collapse of the structure are shown in Figures 6b, c, and d. It is found that a minimum of three hinges are required for the structure to collapse. By considering each of these mechanisms separately and applying the bounding theorems it can be shown that the failure mechanism shown in Fig. 6c, gives the lowest upperbound value [11]. The analytical collapse load has been shown to be [11]:

$$F_L = \frac{M_p}{2 L} \quad (25)$$

Using the GLOSS R-Node method of analysis, and plotting the r-node stresses it is found that the three highest peak values occur at the hinge locations shown in Fig. 6c. The limit load (Table 1) is found to compare well with the analytical results.

A two bay framed structure with loading is shown in Fig. 7. The beams and columns are 10 inch long ($2L$), the beam cross-sections are 2 inch $\times$ 1 inch and the columns are 1 inch $\times$ 1 inch. This structure will collapse when a minimum of seven hinges form. These hinges could form at any of the locations 1 to 10. This structure has been analysed by the method of combining mechanisms [11], and after considering a multiplicity of combinations, the locations for the formation of hinges, leading to the lowest upper bound results, are at 1, 3, 4, 5, 8, 9 and 10. The limit load is found to be [11]:

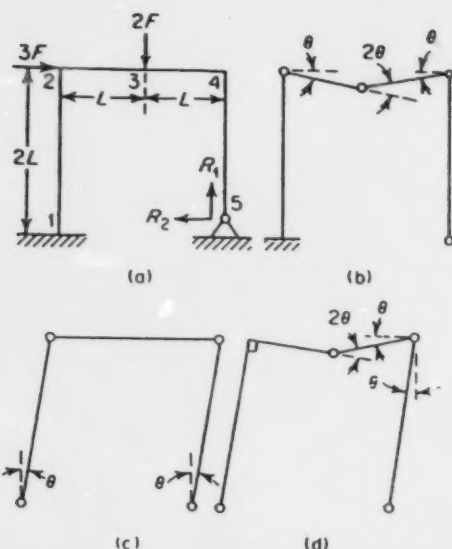


Figure 6 : Single Bay Frame with Loads and Possible Mechanisms

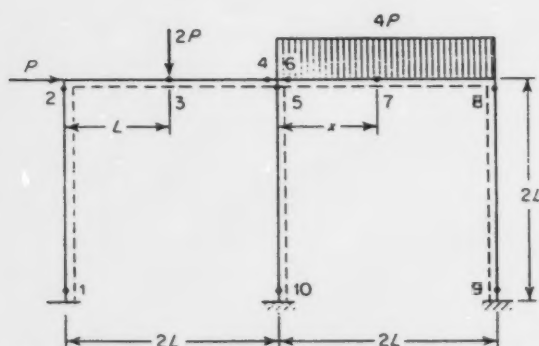


Figure 7 : Two Bay Frame with Concentrated and Distributed Loads

$$P_L = \frac{M_p}{0.364 L} \quad (26)$$

The plot of the r-node stresses for each of the columns and the beams show that the seven peak values occur at locations 1, 3, 4, 5, 8, 9 and 10, this is the same as that obtained from the analytical method and the limit load compares well with the analytical results (Table 1).

Arched Structures (Fig. 8)

The analytical solution for the limit loads of arches is possible through the use of limited interaction curves [2]. This method locates the hinges at the center of the arch and at the two points situated symmetrically about the center at an angle χ and $180 - \chi$ from the horizontal. The value of χ can be found from the solution of the transcendental equation

$$\cos(\phi_0 - \chi) = 1 - \frac{2(1 - \cos \chi)}{1 - h}$$

where ϕ_0 is the semi included angle of the arch. The limit load can be found to be [4]:

$$F_L = \frac{M_p}{A} \frac{1 - h - 2 \cos \phi_0 + (1 + h) \cos(\phi_0 - \chi)}{(1 + h) \sin \chi + (1 - h^2) \sin(\phi_0 - \chi) - (1 - h) \sin \phi_0}$$

where A is the radius of curvature of the arch, $h = \frac{M_p}{A N_p}$, N_p being the collapse load in tension.

The plot of the r-node stresses for an arch with $A = 12.5$ ins., $\phi_0 = 90^\circ$ and cross-section 1 inch $\times$ 1 inch, with a concentrated load at the center shows r-node peak stresses at locations which agree well with the analytical hinge locations, and the limit loads calculated with these peak values show the GLOSS R-Node method to be accurate (Table 1).

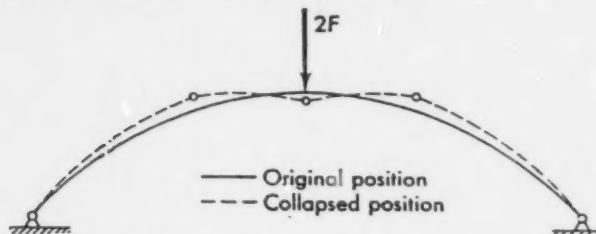


Figure 8 : Plastic Hinges in an Arched Structure

CONCLUSION

Estimates of plastic collapse loads can be obtained using the GLOSS R-Node method. Configurations involving beam structures and framed structures have been analyzed by the method, and the results are encouraging. The GLOSS R-Node method can be extended to more complex configurations involving shear and torsional loads. The ease with which this method can be applied should be attractive to practicing engineers. Further refinement of the method could result in the automatic determination of plastic collapse loads once the configuration and sectional properties are stipulated.

ACKNOWLEDGMENT

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Table 1
COMPARISON OF LIMIT LOAD ESTIMATES

Structure		Collapse Load Analytical (psi.)	Collapse Load GLOSS R-Node (psi.)	Percentage Difference %
1.	(a) Beam fixed at one end and simply supported at the other end	218.6	215.12	-1.6
	(b) Beam fixed at both ends	300.0	304.15	1.4
2.	(a) Single Bay Frame	750.0	766.73	2.2
	(b) Two Bay Frame	4120.9	4068.63	-1.3
3.	Arched Structure $A = 12.5, \phi_0 = 90^\circ$	4740.85	4770.85	0.6

Yield stress = 30,000 psi

TECHNICAL SESSION 8
PLANT LIFE ASSESSMENT

Chair: W.H. Gardiner



INTEGRITY ASSESSMENT OF THICK WALLED INDUSTRIAL FURNACE TUBES

R. K. Kizhatil and R. Seshadri
Industrial Systems Engineering
University of Regina, Regina, Canada.

ABSTRACT

This paper examines various simplified methods proposed to analyze stresses and predict damage and remaining life in furnace tubes subjected to sustained primary pressure stresses and cyclic secondary thermal stresses resulting from a typical furnace operation. Operational effects such as tube fouling, firing rates, startup-shutdown cycles are considered. Component integrity assessments are carried out using some recently developed techniques. A numerical example of a furnace tube made of HP 45 material is presented, and results obtained using a non-linear finite element analysis are compared with predictions obtained using the elastic-core method.

NOMENCLATURE

α	coefficient of thermal expansion
γ	constraint parameter
δ	upper bound on strain increment
$\epsilon^*(t)$	total strain accumulation in time t
ϵ_{cr}	steady-state creep strain
λ	creep magnification factor
σ_{ti}	initial membrane hoop stress
σ_{tp}	primary pressure stress
σ_{tt}	secondary thermal stress
σ_{tli}	maximum thermoelastic hoop stress
σ_c	core stress
σ_s	relaxed steady-state core stress
σ_y	yield stress
a, b	elastic-core boundaries measured from tube mid-wall
B, n	steady-state creep parameters
c	corrosion rate of tube wall
C	Larson-Miller constant
D	creep damage
D_o, D_i	outside and inside diameters
E	Young's modulus
h, h_0	initial tube wall thickness
$h(t)$	tube wall thickness in time t
k	thermal conductivity
L	rupture life at specified stress and temperature
p_r	internal pressure for creep rupture evaluation
p, q	material life estimation parameters
P_{LM}	Larson-Miller parameter
P	primary stress parameter
Q	secondary stress parameter
$r_{e/c}$	radius of elastic-core boundary
S_r	stress to rupture in specified time
t_r	time to rupture under varying stress

t_0	time to rupture under initial membrane hoop stress
T	temperature
ΔT	temperature difference
Y	diameter ratio ($= D_o/D_i$)

INTRODUCTION

The analysis and assessment of remaining life of furnace tubes has been a problem of considerable significance in the petrochemical industry. Typically, these tubes which operate in the creep range of temperature are subjected to complex mechanical and thermal load histories. Consequently their analysis can be very involved and time-consuming. Various simplified methods [1,2,3] for the analysis and life assessment of furnace tubes have been proposed, some of which are examined in this paper.

Furnace tubes operating at elevated temperatures experience creep damage and subsequently fail due to the formation of fissures or creep voids. As creep damage accumulates with time, it becomes necessary at some point to evaluate the remaining life in order to make decisions regarding tube retirement. The amount of creep damage that has occurred may be calculated by examining the time-temperature-stress history of the furnace tube. The fraction of the tube-life expended for each time-temperature segment of the history is calculated using the life estimation parameters of the material and summed up using a life-fraction rule. While this procedure is known to produce inaccurate results due to errors in the operating history, scatter in the material data and the approximate nature of the life-fraction rule, it is, however, extremely useful in determining whether or not a more detailed evaluation is required and in identifying critical locations that undergo substantial creep damage. Very often a history-based life assessment is followed up by a direct evaluation involving non-destructive examination.

TUBE DESIGN AND INTEGRITY ASSESSMENT METHODS

Methods Based on the Mean Diameter Formula

The American Petroleum Institute's Recommended Practice RP 530 method [1] for the design and life prediction of heater tubes is based on the use of the "mean diameter equation". A constant pressure, p_r , based on steady state operating conditions is assumed. Thermal stresses are neglected and a membrane hoop stress based on the mean diameter of the furnace tube is calculated as

$$S_r = \frac{p_r [D_o - h]}{h} \quad (1)$$

Using the maximum outside wall temperature, the rupture life at the stress S_r for the furnace tube material tube is calculated. The choice of the mean diameter location for rupture damage assessments has been the subject of discussion of several papers.

Life predicted by the mean diameter formula has been known to be far in excess of operating experience. This is due to the interaction of the primary pressure stress in the tube with the cyclic secondary thermal stress during actual operation that would lead to considerably larger damage than that predicted by the API method.

Finnie [2] has presented a graphical method of calculating damage in furnace tubes. In order to use this method to the best advantage an accurate estimate of the rate at which the furnace tube corrodes at the actual operating conditions should be known. This may be determined experimentally, or actually measured on an operating furnace tube at discrete intervals.

The rupture life, L , at a stress σ and the temperature T is given by the familiar empirical relation

$$L = p\sigma^{-q} \quad (2)$$

where $p = p(T)$ and $q = q(T)$ are the material life estimation parameters which are functions of temperature. Creep rupture is assumed to occur when

$$\int_0^{t_r} \frac{dt}{L} = 1 \quad (3)$$

where t_r is the time to rupture.

If the corrosion rate, c , is uniform, then the wall thickness at any time t is given by

$$h(t) = h_0 - ct \quad (4)$$

where h_0 is the initial wall thickness. Using equations (1) and (3) in (2), Finnie obtained the following expressions

$$\left. \begin{aligned} t_r/t_0 &= 1 - (1 + [q-1]\beta)^{1/1-q} & q \neq 1 \\ t_r/t_0 &= (1 - \exp^{-\beta})/\beta & q = 1 \end{aligned} \right\} \quad (5)$$

in which $\beta = ct_0/h_0$, and t_0 is the rupture time at an initial membrane hoop stress of $\sigma_{\theta i} = pR_o/h_0$. For a given furnace tube, the initial membrane hoop stress $\sigma_{\theta i}$, the rupture parameters $p(T)$ and $q(T)$, and the time to rupture $t_0 = p(\sigma_{\theta i})^{-q}$ are first evaluated. Then using Eqn. (4), the time to rupture for the furnace tube, t_r , is obtained. Finnie presented Eqn. (4) in a graphical form as plots of t_r/t_0 against $\beta = ct_0/h_0$ for various values of q for ease of computation. The time to rupture t_r is then obtained from the plots of t_r/t_0 against $\beta = ct_0/h_0$.

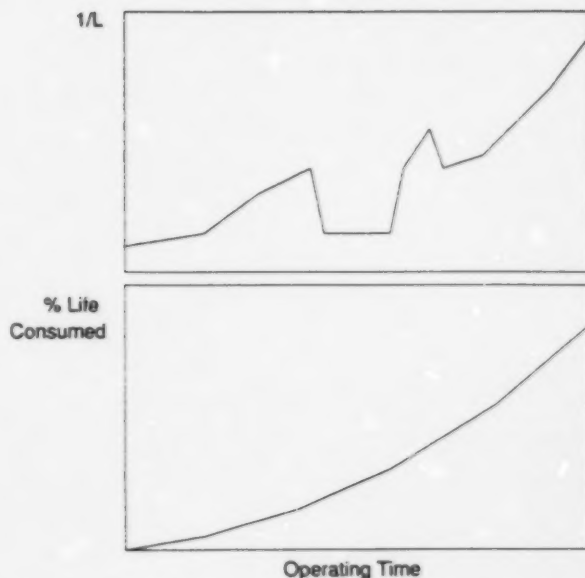


Fig. 1 : Graphical Presentation of Cumulative Life Calculation.

For those situations in which the corrosion rate, pressure, temperature and wall thickness continuously vary with time, these variables are measured at discrete intervals and assumed to have remained steady between any two measurements. The rupture life $L = p\sigma^{-q}$ is calculated based on this assumption and $1/L$ is plotted against operating time as shown in Fig. 1. As Eqn. (2) implies, the area under the $1/L$ versus operating time curve gives the amount of damage incurred at any instant using which the remaining life is assessed. Fig. 1 also shows the damage calculated from the area under the $1/L$ versus operating time curve. When the damage reaches unity, the life of the tube is expended and it would have to be replaced.

In the above calculations thermal stresses have been neglected because it has been assumed that they relax fairly rapidly due to creep. However it has to be recognized that thermal stresses can be significant in situations involving high heat flux, thick-walled tubes, low thermal conductivity and cyclic operation. The elastic-core method, discussed next, may be used in such situations as it takes into account the interaction between the primary pressure stress and the secondary thermal stress. Also, the maximum tensile stress $\sigma_{\theta} = pR/h$ is used in calculating damage and rupture life rather than the Mises equivalent. This has shown to be acceptable and even to predict better results for materials that develop fissures during creep at elevated temperatures.

The Elastic-Core Method [3]

Bree [4] performed a simplified analysis of a thin-walled tube subjected to a combination of a steady primary pressure stress and a cyclic thermal secondary stress, and identified regions of shakedown (S_1 and S_2), alternate plasticity (P) and ratcheting (R_1 and R_2) on an interaction diagram, Fig. 2. The interaction diagram is a plot of the secondary thermal stress $\sigma_{\theta t}$ against the primary stress $\sigma_{\theta p}$ as in Fig. 2 or of the secondary stress parameter $Q = \sigma_{\theta t}/\sigma_p$ against the primary stress parameter

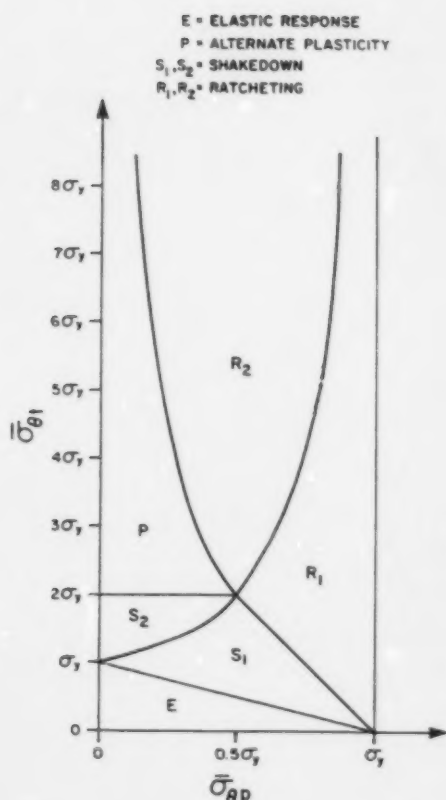


Fig. 2: The Interaction Diagram.

$P = \sigma_p/\sigma_y$ as in Fig. 3. Frederick and Armstrong [5] have shown that repeated application of a loading cycle on a structure will modify the stresses in the structure so that they will converge to a steady cyclic state of stress. For thin-walled tubes, O'Donnell and Porowski [6] have shown that if the cyclic operating conditions are in the shakedown (S_1 and S_2) or alternate plasticity (P) regions, then the steady-state stress distribution developed after some startup-shutdown cycles will consist of an elastic region called the "elastic-core". Seshadri [3] has modified the elastic-core method for thick-walled furnace tubes taking into consideration the actual stress distribution in thick tubes due to pressure and thermal loads. In this paper, the elastic-core method is applied to a thick-walled furnace tube.

The elastic-core stress σ_c is obtained from equilibrium considerations and may be calculated for different stress regions from Table 1 or directly from the interaction diagram, Fig. 3, as

$$\sigma_c = \lambda \sigma_p \quad (6)$$

where λ is the creep magnification factor and σ_p is the primary pressure stress. The elastic-core stress σ_c relaxes to a steady state value σ_s such that $\sigma_c > \sigma_s \geq \sigma_p$ during a high temperature operating cycle. Creep damage occurs during an operating cycle as the elastic-core stress relaxes from σ_c to σ_s and also at the steady state stress level σ_s . If it is conservatively assumed that the maximum elastic-core stress σ_c does not relax, then the rupture life at the operating temperature for any i th cycle may be obtained as $(t_r)_i = p\sigma_c^{-q}$ so that the damage in each operating cycle of hold time t_i may be summed to obtain the total damage

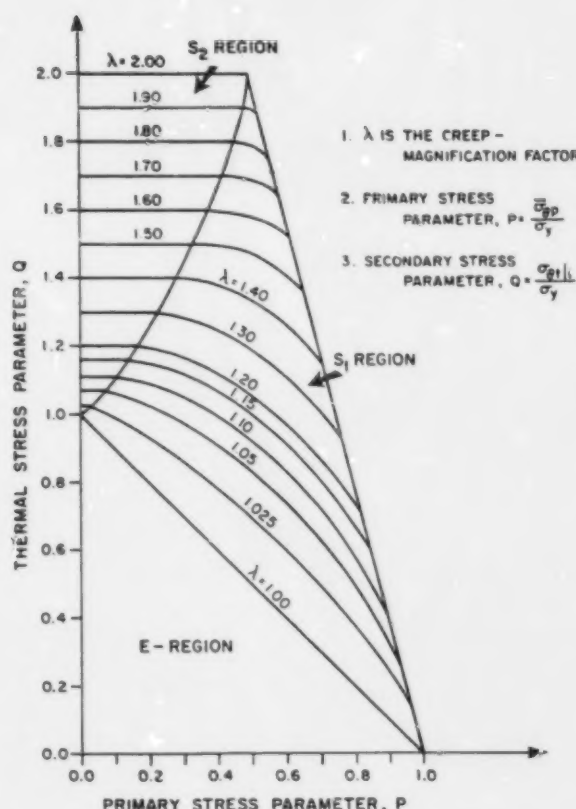


Fig. 3: The Interaction Diagram With the Creep Magnification Factor, λ .

as

$$D = \sum_i \frac{t_i}{(t_r)_i} \quad (7)$$

and the life-fraction rule used to predict remaining life.

Realistically the maximum elastic-core stress relaxes with time so that the damage computed by Eqn. (7) would be conservative. The GLOSS method of analysis [7] can be used to estimate stress relaxation as

$$\sigma(t) = \frac{1}{[\sigma_c^{1-n} + \gamma BE(n-1)t]^{1/(n-1)}} \quad (8)$$

where γ is the constraint parameter, σ_c is the initial core stress, $\dot{\epsilon}_c = B\sigma_c^n$ is the steady state creep relationship and $\sigma(t)$ gives the relaxed core stress at any time t . Using Eqn. (8) in the expression for damage; i.e.,

$$D = \int_0^t \frac{dt}{L} \quad (9)$$

where $L = p\sigma^{-q}$ is the rupture time leads to the expression for damage D as

$$D = \frac{1}{(q+1-n)pBE} [\sigma_c^{q+1-n} - \{\sigma_c^{1-n} + \gamma BE(n-1)t\}^{(q+1-n)/n}] \quad (10)$$

Eqn. (10) gives the damage incurred in time t taking into account the relaxation of the maximum elastic-core stress. The constraint parameter γ which appears in Eqn. (8) and (10) is

obtained using a GLOSS analysis and it characterizes the multiaxiality and geometric constraint at the location under consideration.

The elastic-core method can also be used to bound the total strain accumulation as

$$\epsilon^*(t) = \frac{\sqrt{3}}{2} \left\{ \sum_i \epsilon_{cr} + \sum_i \delta \right\} \quad (11)$$

where $\epsilon_{cr} = B\sigma_s^m \Delta t_i$ is the steady creep strain incurred at the relaxed stress level σ_s for a time Δt_i , and

$$\delta < \frac{1}{E} \left\{ \frac{\sigma_i^2 - \sigma_s^2}{\sigma_s} \right\} \quad (12)$$

is a bound on the strain increment incurred as the maximum elastic-core stress relaxes from σ_i to σ_s . The factor $\sqrt{3}/2$ in Eqn. (11) is a correction for the multiaxial stress state in the tube.

OPERATIONAL FACTORS AFFECTING TUBE DESIGN

Firing Rate and Tube-Metal Temperature

The temperature gradient set up across the thickness of a furnace tube is related to the average radiant heat flux, q_o , tube dimensions, D_o and $Y = D_o/D_i$, and thermal conductivity of the tube material, k , as

$$\Delta T = \frac{q_o D_o}{2k} \ln Y \quad (13)$$

The average heat flux and the tube-metal temperature have to be determined for the purposes of design and life assessment by any of the previously described methods. Tube-metal temperatures cannot be estimated with sufficient accuracy for design and life assessment calculations due to random variations. This can lead to considerable error in design as rupture life and allowable stresses are very sensitive to temperature. The API RP 530 design method introduces a safety factor by using a conservative estimate of the outside wall (maximum) temperature, usually 50 deg F above the maximum operating temperature in order to allow for possible overheating of tubes.

Tube-Wall Thickness

For given operating conditions and tube diameter, the tube-wall thickness has to be chosen so as to minimize the value of the creep magnification factor λ in Fig. 3. Increasing the tube wall thickness reduces the value of P and increases Q from the definition of these parameters. An optimum tube-wall thickness leads to a minimum value of λ and of σ_c and hence minimum damage and maximum tube life.

Tube Fouling

The formation of scales and deposits on the tube surface can affect tube heat transfer and lead to higher tube-metal temperature. This leads to increased damage and strain accumulation.

Start-up and Shut-down Cycles

Eqns. (7) or (10) may be used to determine the damage accumulation and strain increment during each operating cycle. In

order to use these equations for tube design some estimate of the sequence of start-up and shut-down cycles should be available.

NUMERICAL EXAMPLE

A thick-walled furnace tube of inside diameter 4.063 in (103.22 mm), outside diameter 5.563 in (141.3mm), sustained internal pressure 425 psi (2.93 MPa), inside wall temperature 1350 deg F (843.33 deg C) and outside wall temperature 1550 deg F (843.33 deg C) is considered. The tube is subjected to a cyclic thermal gradient. The tube material is HP 45 alloy, and material properties are evaluated at a mean temperature of 1450 deg F (787.77 deg C). Calculations based on the mean diameter formula and the elastic-core method are presented in this section. The results obtained are also validated using nonlinear finite element analysis.

Mean Diameter Formula Method

Using Eqn. (1), the stress based on the mean diameter formula can be calculated as 1363.68 psi. A conservative value for the operating temperature would be 1600 deg F, which is 50 deg F more than the maximum outside wall temperature. The rupture life for HP 45 alloy at 1600 deg F and 1363.68 psi is estimated to be 9.098×10^5 hours. From a knowledge of the sequence of start-up and shut-down cycles, the amount of creep damage accumulated can be calculated using Eqn. (7).

Finnie's method can be used if the corrosion rate of the furnace tube in its operating environment is known. The initial membrane hoop stress is $\sigma_{\theta i} = 1576.18$ psi, and again assuming a conservative temperature of 1600 deg F, the time to rupture at this stress is $t_0 = 5.54 \times 10^5$ hours. Assuming a corrosion rate of $c = 0.02$ in/10,000 hours, the time to rupture is calculated using Eqn. (4) as $t_r = 2.59 \times 10^5$ hours.

Elastic-Core Method

The elastic-core method [3] which takes into consideration the interaction between the primary and secondary stresses in thick-walled tubes can be used to assess damage and remaining life in the following manner:

The primary stress parameter is

$$P = \frac{\sigma_{\theta p}}{\sigma_s} = \frac{pR_i/(R_o - R_i)}{\sigma_s} = 0.057$$

and the secondary stress parameter is

$$Q = \frac{\sigma_{\theta i}}{\sigma_s} = \frac{E\alpha\Delta T}{2(1-\nu)\ln Y} \left[\frac{2Y^2}{Y^2-1} \ln Y - 1 \right] \frac{1}{\sigma_s} = 1.5773$$

where $\sigma_{\theta p}$ is a pressure induced membrane hoop stress and $\sigma_{\theta i}$ is the maximum thermoelastic hoop stress which occurs at the inside bore. From Fig. 3 the stresses are seen to be in the S_2 region, and $\lambda = Q = 1.58$. The maximum elastic-core stress is

$$\sigma_{sc} = \lambda\sigma_{\theta p} = 1816.00 \text{ psi}$$

To calculate creep damage or rupture life corresponding to the value of the maximum elastic-core stress, the rupture parameters p and q have to be determined at the appropriate temperature. From Table 1 the elastic-core extends to a radius of $r_{e/c} = 2.62$ inches. The temperature at this radius is obtained

Table 1
Elastic-core stress and core thickness

Stress regime	Elastic-core stress σ_c	Elastic-core boundary ($x = a$)	Elastic-core boundary ($x = b$)
S_1	$(\sigma_{s1} - \sigma_y) - 2\sqrt{\sigma_{s1}(\sigma_y - \sigma_{sy})}$	$\{1 - 2\sqrt{(\sigma_y - \sigma_{sy})/\sigma_{sy}}\} (h/2)$	$(h/2)$
S_2	$(\sigma_{sy}\sigma_{s1})/\sigma_y$	$\{(\sigma_{sy}/\sigma_y) - (\sigma_y/\sigma_{s1})\} (h/2)$	$\{(\sigma_{sy}/\sigma_y) + (\sigma_y/\sigma_{s1})\} (h/2)$

Notes:

1. σ_c is the maximum stress in the elastic-core which exists after start-up.
2. During the fired heater operational period, the core stress can relax to a stress level σ_s , such that $\sigma_{sy} < \sigma_s < \sigma_c$.

from the logarithmic temperature profile as $T_{s/c} = 1512.63$ deg F.

Using the Larson-Miller parameter and the creep rupture properties of HP 45 alloy, the rupture parameters p and q are evaluated at different temperatures as shown in Table 2. The Larson-Miller parameter is given by

$$P_{LM} = (T + 460)(C + \log t) \quad (14)$$

where

- T = temperature in deg F
 t = rupture time at $T^\circ\text{F}$ and stress level σ
 C = Larson-Miller constant = 19.52 for HP 45 alloy

Experimental data regarding the times to rupture, t , at specific temperatures T and stress levels σ are available. From these the time to rupture at any required temperature, such as the temperature at the elastic-core boundary, can be calculated using Eqn. (14). By evaluating the times to rupture at a given stress level for two different temperatures, and using $L = p\sigma^{-q}$, the values of the life estimation parameters p and q are calculated. Table 2 gives the values of p and q at temperatures of interest obtained by this procedure. A more refined method to calculate p and q is to use some form of regression analysis of experimental data on rupture to obtain a best fit equation [8]. However, in this paper, for the sake of simplicity the above described procedure is used to calculate the life estimation parameters.

If it is conservatively assumed that the maximum elastic-core stress does not relax significantly, as is the case for some of the highly creep resistant furnace tube materials such as HK 40 and HP 45, then the amount of damage in, say, 50000 hrs

of operation at the inside bore and at the elastic-core boundary may be calculated using

$$D_i = \frac{t_i}{(t_r)_i}$$

Table 3 shows the temperature, stress level and the damage incurred in 50,000 hrs of operation at both the elastic-core boundary and the inside bore.

From Table 3, it can be seen that damage is higher at the elastic-core boundary than at the inner-bore. Alternately, Eqn. (10) may be used if stress relaxation is taken into account. GLOSS analyses are performed to obtain the constraint parameter γ in Eq. (10) at the elastic-core boundary ($\gamma = 0.8755$) and at the inside bore ($\gamma = 0.8754$). Details of performing a GLOSS analysis have been reported elsewhere [7,9]. Damage obtained using Eqn. (10) at the elastic-core boundary and the inside bore are plotted in Fig. 4, which again shows higher damage at the elastic-core boundary. These observations are in agreement qualitatively with results obtained using a detailed computational model by Simonen and Jaake [10]. The elastic-core boundary has a lower stress level but a higher temperature compared to the inside bore which has a higher stress level but lower temperature. Rupture life and hence damage are functions of both stress and temperature, and in this case the higher temperature causes more damage. Creep damage reaches unity, and fissuring occurs first at the elastic-core boundary. Fissuring then spreads both outwards and inwards from the elastic-core boundary, and reaches the inner-core before reaching the outer surface of the tube. These computational predictions are in keeping with metallographic examinations of retired tubes.

Validation using Inelastic Finite Element Analysis

A non-linear elastic-plastic-creep finite element analysis of the furnace tube problem is also performed using the ANSYS [11] finite element software. Axi-symmetric isoparametric quadrilateral STIF 42 elements are used to model the furnace tube. Nodal coupling is carried out to prevent bending due to the thermal gradient. The furnace tube is subjected to a sustained internal pressure of 425 psi and a cyclic thermal gradient of 200 deg F. The inside bore is maintained at 1350 deg F and the outside tube-metal-temperature at 1550 deg F during furnace operation. After start-up, the tube is subjected to the sustained pressure

Table 2
Life Estimation Parameters

Temperature deg F	p	q
1350.00	4.767×10^{21}	3.892
1512.63	1.913×10^{18}	3.571

Table 3
Comparison of Damage at the elastic-core boundary
and inside bore in 50,000 hours

Elastic-core boundary	
Temperature, $T = 1512.63$ F	
Stress, $\sigma = 1815.99$ psi	
Damage, $D = 0.0113$	
Inside bore	
Temperature, $T = 1350.00$ F	
Stress, $\sigma = 7242.26$ psi	
Damage, $D = 0.0108$	

and the thermal gradient for a period of 50,000 hrs at which time the thermal load is removed (shut-down). For the second cycle the thermal load is once again applied for a period of 100,000 hrs and then shut-down occurs. A third cycle of 100,000 hrs is then imposed. Material properties are taken same as in the elastic-core method at 1450 deg F.

Fig. 5 shows the hoop stress distribution across the thickness of the tube in the third loading cycle. Fig. 6 is the stress distribution predicted by Bree's [3] simplified analysis which can be seen to be in good agreement with the actual distribution in spite of the effect of curvature and thickness effects in the thick-walled tube considered. The finite element analysis implicitly takes into account the effect of curvature unlike the assumptions in the Bree model. It is also observed that the hoop stress distributions obtained using the finite element analysis for the second and third operational cycles are almost identical thus verifying the theorem of convergent internal stresses [5]. The existence of an elastic-core can also be seen in Fig. 5. The elastic-core stress actually increases slightly at first immediately after start-up, and then relaxes. The maximum elastic-core stress during an operational cycle by the finite element analysis is $\sigma_{sc} = 1762.44$ psi. The elastic-core method predicts a value of 1815.99 psi for the maximum elastic-core stress which leads to conservative estimates of damage, remaining life and tube growth.

A furnace tube of the same dimensions but with inside bore temperature of 1400.5 deg F (760 deg C) and an outside tube-metal temperature of 1500 deg F (816 deg C) is also analyzed. The parameters P and Q are such that the stresses are in the elastic E regime in Fig. 3. Damage calculations show that damage is higher at the inner bore as compared to the elastic-core boundary. This might explain the initiation of damage having been observed at the inside bore in some furnace tubes in actual operation. Obviously such tubes are overdesigned for their operating conditions since they operate in the E stress regime. This example also shows that errors in estimating the tube-metal temperature can lead to vastly different predictions regarding the inelastic response, damage and remaining life.

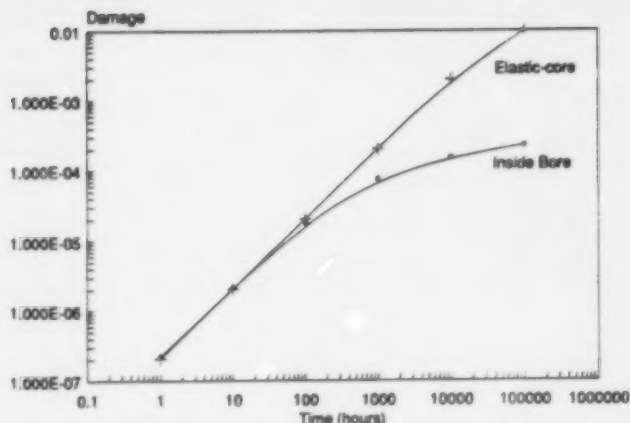


Fig. 4 : Creep Damage at the Elastic-Core Boundary and Inner-bore.

PRACTICAL CONSIDERATIONS IN TUBE LIFE ASSESSMENT AND RETIREMENT

Industrial plant operators are often faced with the decision of retiring those furnace tubes which may not survive a forthcoming operational cycle. Creep damage can be calculated using analytical methods such as the elastic-core method or using a detailed finite element analysis so as to determine whether a tube needs to be replaced. However, plant operators are faced with some practical problems in implementing any of the previously described methods. These include lack of reliable data on material properties, difficulty in the measurement of tube-metal temperature and heat flux, and inaccuracies in the time-temperature history. Also, the option of performing a non-linear finite element analysis may prove to be impractical, expensive and time-consuming in such an environment. Consequently, plant operators resort to other direct methods for estimating remaining life [7], some of which are

1. Measurement of diametral strain which works well only for wrought alloys that expand significantly before failure. Cast furnace tubes of HP 45 or HK 40 alloys show less than 1% expansion prior to failure.
2. Non-destructive examination such as surface examination, ultrasonic, magnetic and radiographic examinations.
3. Creep rupture testing of material from damaged furnace tubes.
4. Metallographic examination.

The analytical and computational methods described in the previous section may be used by plant operators to determine which tubes are at the end of their useful life. These tubes may then be subjected to one or more of the above methods of direct evaluation to confirm the theoretical damage predictions. While the above mentioned direct methods are useful to determine whether a given furnace tube needs to be replaced or not, they do not, in general, provide a quantitative estimate of the remaining life of those tubes that are not replaced even if the forthcoming operational cycles are known. For these reasons, plant operators have to use a judicious combination of analytical/computational methods and direct evaluation methods before making expensive tube replacement decisions.

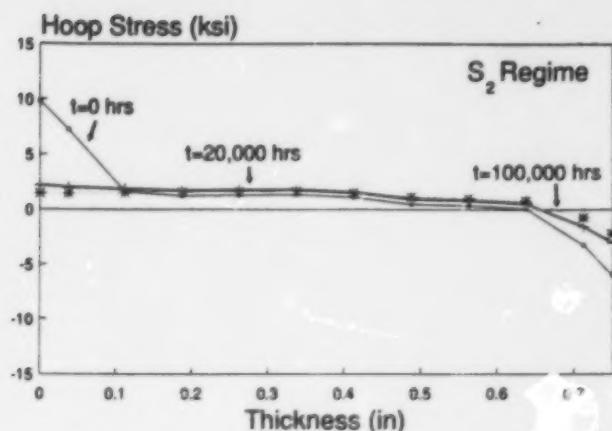


Fig. 5 : Hoop Stress Distribution for Steady State Cycles (Inelastic FEM).

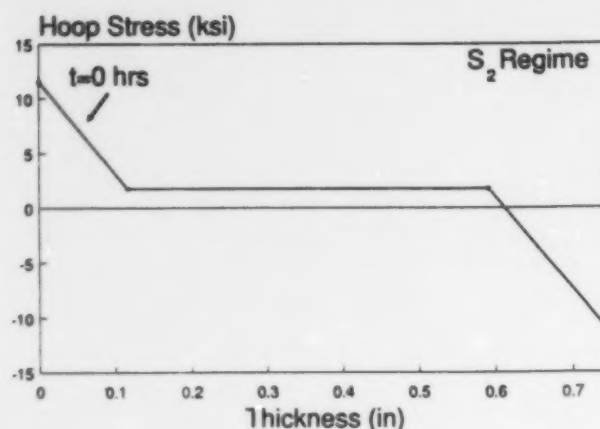


Fig. 6 : Hoop Stress Distribution for Steady State Cycles (Elastic-Core Method).

CONCLUSIONS

Analytical procedures for the design and life prediction of furnace tubes are presented in this paper. The API RP 530 method based on the mean diameter equation is known to sometimes imply tube life far in excess of operating experience. Finnie's method neglects thermal stresses, and can be used only when the rate at which a furnace tube corrodes is known. The elastic-core method takes into account the non-linear interaction between the cyclic thermal stresses and the sustained pressure stress. Computational methods, such as finite element analysis, are expensive and time-consuming, but may be used if a detailed analysis is justified. A numerical example of a furnace tube is presented, and results predicted by the elastic-core method are found to be conservative compared to the finite element method. It is suggested that tube retirement decisions be made on the basis of damage calculations using analytical or computational methods in conjunction with existing direct methods such as measurement of tube growth, NDE, creep rupture testing and metallographic examination.

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REMAINING LIFE ASSESSMENT - MAKING A START

by

W.H. Gardiner
Manager, Life Extension
SaskPower

ABSTRACT

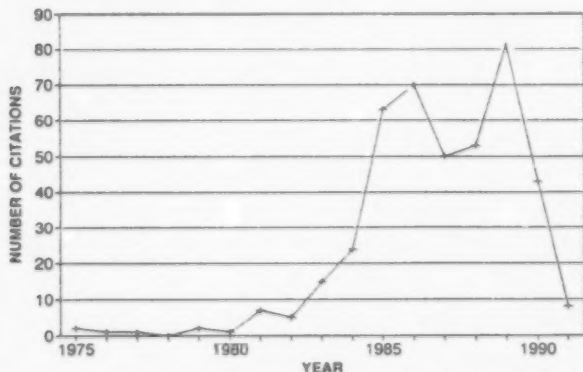
Life extension of major plant is a relatively recent requirement of many company plans. This is frequently necessary to reduce corporate capital requirements in times of slow economic growth.

Most of the remaining life assessment required for a project, can be accomplished with existing company expertise, thus leaving only the critical assessments and difficult decisions to the specialist materials experts.

A brief review of methods and procedures is included.

1. INTRODUCTION (A brief review of Life Extension)

Life Extension appears to be a process that has suddenly "caught up" with us. References to Life Extension in the literature began to increase in the early 1980's and have currently settled down to a healthy flow, see Figure 1. (1)



"LIFE EXTENSION" CITATIONS
IN ENGINEERING INFO. INC.

FIG. ONE

Until the mid 1980's, the management of most companies looked favourably on the purchase of new plant. The new equipment, was usually bigger and better than the equipment it replaced. It might also need less people or less energy. Large capital expenditure was almost welcome; it implied expansion and faith in the future. But that was when the economy was growing.

Now in the 90's with a slow economy and many companies showing dangerously low returns, the management of many companies does not need to expand capacity and prefers not to replace aging equipment. They would much prefer to make the equipment last a few more years.

It is currently not unusual for the financial planning arm of a company to ask the engineering group whether plant can indeed be life extended, as a cost control measure.

In the 1990's, plant which is currently aging, is often technologically much closer to the current design and performance, than was the post war equipment which was so readily discarded in the 70's and 80's.

The preference is now to postpone retirement of equipment rather than replace it, possibly prematurely. It can be stated that the pressures which are going to drive Life Extension in the 90's are financial necessity and the greater likelihood of continuing mechanical suitability of the aging equipment.

So What is Life Extension?

The term Life Extension can be defined in a number of ways:

1. Life Extension - The Creed of the Maintenance Manager

'To prolong the service life of plants or components, with no loss of plant reliability, by optimum maintenance and operation practices.'

(This is no more than any maintenance manager or planner would consider his daily objective.)

2. Life Extension - Where unreliability carries a small penalty.

'To continue operation after the expected retirement date with minimum assessment of the likelihood of component failure. To have contingency plans for periods of non-production.'

(This may be acceptable to an organization which can rationalize the occasional complete shutdown and where failure is not a safety hazard, but may prove no less expensive than option 3.)

3. Life Extension - Where unavailability carries a high penalty.

To postpone the retirement of a plant or component at the end of its traditional life, with no loss of plant availability, by pro-active assessment and where necessary upgrade of equipment.

(An option being considered much more in the current times of economic retrenchment as an alternative to plant replacement.)

I don't think that definition #1 needs much further mention. Clear differences in objective and scope were seen as early as 1986,⁽²⁾ but the differences between definitions #2 and #3 deserve some discussion.

The first difference is the required reliability of the operation. Number 3 implies a high level of output and safety. For instance, organizations and equipment that transport people - airlines, building elevators, etc., must opt for #3. Additionally, where the actual failure of the equipment may involve personal injury, e.g. pressure vessels, high speed rotating equipment, etc., again #3 is the only credible option.

Definition #2 is only tolerable for industries which can accept breakdown, and to whom the "lost production costs" during shutdown are acceptable. Few industries can say "Wait until it breaks and then we'll fix it." Much of the difference between #2 and #3 involves the philosophy of proactive versus reactive thinking.

The number of industries with an interest in life extension is increasing. The number of opportunities to postpone new plant construction increases daily. We can substantially reduce capital investment by extending operation of aging plant. We must be reasonably sure that overall plant performance and reliability will be maintained.

With a good grasp of the various stages of the assessment and decision making process, many organizations can progress much, if not all of the way, with in-house resources, to perform the needed examinations to determine whether a substantial extension of life is possible.

2. PREPARING FOR THE ASSESSMENT

The usual Life Extension Project comprises a number of steps, most of which are indispensable. (See Figure 2.)

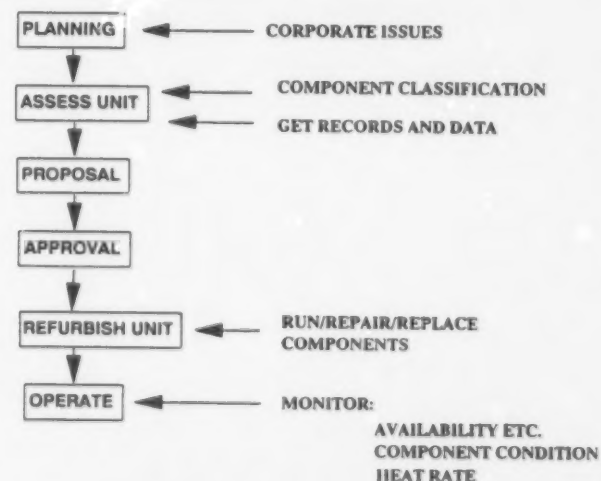


FIGURE TWO
STEPS OF PLANT LIFE EXTENSION

The most difficult step of the process to organize is the assessment of remaining life of the plant or system. Failure to produce an accurate and cost effective remaining life assessment will affect the usefulness of the whole Life Extension process.

The first intention to life extend, will usually have come from a corporate planning or financial group. This intention should be regarded as an option at this stage, until the remaining life assessment has been undertaken and accepted as the preferred option to meet future needs.

It is desirable that this assessment begin with some research into:

1. Corporate objectives and options.
2. The role that the plant or system in the proposal currently performs within the company.
3. A comparison of roles of any competing or supporting plants/systems.
4. Any changes of input/output in the new role the plant will play, during life extension.
5. The ranking of the benefits to the company, of all the options.

This process clears the perception about the importance of the plant or system to the organization, and confirms what is truly the first option to evaluate. This will allow the resolution of any differences in perception at an early stage.

3. ASSESSMENT OF REMAINING LIFE

3.1 Classification of Components

In order to get started, two questions immediately arise:

1. Which components do we test?
2. How exhaustively do we examine them?

The answer to these and many other questions can be found in 'Generic Guidelines for the Life Extension of Fossil Fuel Power Plants' ⁽³⁾, published by the Electric Power Research Institute, of Palo Alto, California.

This 1986 work combines techniques from Europe and North America with original work, to produce a detailed procedure, such that any organization can carry out much, if not all, of its life extension work, if expertise and manpower is sufficient.

Though the focus is on thermal power plant equipment, the principles are universal.

Components, or groups of components, within the plant or system are first classified as either CRITICAL or INFLUENCE (Non Critical).

All areas must be included in this process.

Definitions of CRITICAL components are typically:

1. Those whose failure would affect worker or general public safety.
2. Those whose failure would result in a prolonged plant shutdown.
3. Those whose replacement would be expensive or require a long shutdown.
4. Those whose replacement part would require a long lead time before delivery.
5. Those which affect plant environmental compliance performance.

The inclusion of a component as critical will vary with each plant.

Where a component or system meets none of the above criteria, it is not likely to be critical. As such it should need lower level examination because the consequences of failure are less severe.

Another possible group of components would include those components that cannot be replaced. These are not expendable and require a thorough assessment of remaining life. Their condition would also have a strong influence on the likelihood of life extension for the whole plant or system.

The non critical equipment usually includes systems with some redundancy or back up. In addition, systems which only have to operate intermittently or can be repaired in a few hours do not warrant major expenditures of time and effort for inspection.

This EPRI procedure ensures that all components are included in the assessment, but only the important ones are examined exhaustively.

3.2 The Assessment

Once we have catalogued and classified all the components of the system, the assessment can begin.

EPRI suggest procedures to ensure that only enough effort is used to obtain the required level of assurance. They suggest a three level approach. (See Figure 3.)

	LEVEL 1	LEVEL 2	LEVEL 3
FEATURE	LEAST DETAIL	MORE DETAIL	MOST DETAIL
FAILURE HISTORY	PLANT RECORDS	PLANT RECORDS	PLANT RECORDS
DIMENSIONS	DESIGN OR NOMINAL	MEASURED OR NOMINAL	MEASURED
CONDITION	RECORDS OR NOMINAL	INSPECTION	DETAILED INSPECTION
TEMPERATURE AND PRESSURE	DESIGN OR OPERATIONAL	OPERATIONAL OR MEASURED	MEASURED
STRESSES	DESIGN OR OPERATIONAL	SIMPLE CALCULATION	REFINED ANALYSIS
MATERIAL PROPERTIES	MINIMUM	MINIMUM	ACTUAL MATERIAL
MATERIAL SAMPLES REQUIRED?	NO	NO	YES

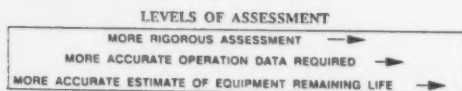


FIG. THREE
(FROM EPRI GENERIC GUIDELINES)

These steps ensure that all the equipment is considered, and that only the marginally acceptable equipment will be exhaustively examined.

The examination involves increasing complexity as a higher level is attempted.

LEVEL ONE

The first level involves application of the appropriate design formula to the minimum material properties of the component. The most pessimistic design or recorded operating parameters are used to assess the amount of life consumed and hence the amount of remaining life available.

A component cannot be subjected to this level of assessment if:

1. Operating records indicate that conditions have exceeded design,
2. The operating history or maintenance files indicate a history of failures,
3. The extended life would involve undue additional stresses,
4. Insufficient operating records exist.

Level one analysis usually confirms suitability of components with conservative original design. Where the required information was not available or sufficient assurance was not obtained, level two analysis will often supply the required confidence in future operation.

Ideally, a large proportion of the critical components should be confirmed as having sufficient remaining life before the level two stage.

LEVEL TWO

This stage uses real operating measurements, minimum material properties, actual dimensions and a visual inspection.

Where no visual defects are noted, the calculations of level one can be repeated using the better data. A further group of components can usually be assured at this stage. This is due to sufficient information now being available that was missing at the level one stage, but also due to better information being available by direct measurements, which will make some calculations of life already expended less conservative.

LEVEL THREE

This stage can involve:

1. Non destructive testing,
2. Material sampling to obtain actual material properties,
3. Strain measurements at key locations,
4. Microstructure evaluation at areas of worst damage,
5. And finally, considerably higher level calculation.

Figure 4 shows some of the techniques available.

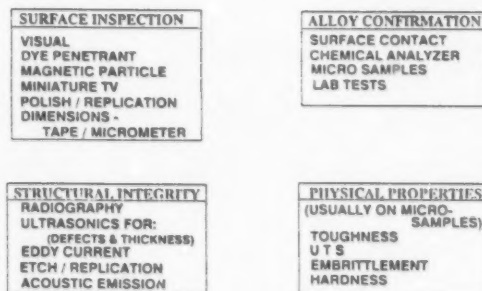


FIG. FOUR
TECHNIQUES AVAILABLE FOR COMPONENT CONDITION ASSESSMENT

If this level of analysis does not assure enough remaining life, some success might be gained with even greater expertise, but this would depend on the worth of the component.

Even when a component is visibly damaged, i.e. by a small crack, fracture mechanics and some study of crack propagation rates can still define some safe remaining life. This will often allow time for later retesting or replacement before the calculated life's end.

This is true for pressure vessels where the configuration implies a 'leak before break' mode of failure rather than something more catastrophic.

It is also quite possible that the conservative nature of the crack propagation calculations, will allow a component to operate subject to future retests. At these retests, the revised remaining life calculation may allow a much longer life in total than the initial prediction.

As can be imagined, it is within this third stage that the most effort and expenditure can occur. It is also at this stage that, depending on the defects uncovered, experts from outside the organization will most likely be required to resolve the remaining life questions.

3.3 The Proposal

The end product of the assessment stage should be a proposal which will contain:

1. A list of the work needed and future testing requirements,
2. The costs for these items.

In compiling this list, a review of all the components and systems (critical and non critical) which are not suitable for the extension period would be made to decide whether to:

RUN REFURBISH REPLACE.

These choices need some further explanation.

RUN

For example if a 15 year life extension was planned for a plant, but a small pressure vessel had only a ten year remaining life (by the most conservative calculations) it would be reasonable to continue operation with biannual inspections of this vessel, if the mode of failure would be benign.

If after some years of operation the damage reached unacceptable levels, a replacement would then be needed. However, if the damage progressed at a slower than anticipated pace, the plant might fulfill the life extension goal without the need for a replacement pressure vessel.

Often the high level of experience and expertise needed to make these decisions can only be found in a few organizations that are extremely specialized in this sphere.

REFURBISH

Many of the systems and components which show signs of wear can be restored by a growing array of specialist companies.

They 'reverse engineer' minor components, often obsolete or of long delivery time. They can often restore items and usually make them more resistant to wear or corrosion than the original piece.

REPLACEMENT

In only a few cases is this option necessary, where technology does not allow the previous repair option. More modern materials may improve component performance. For instance inclusion of some of the new creep resistant steels (ASTM/ASME Grade 91 - 9 % CrMo) in replacement components, might improve the capability of a boiler or heater system.

Opportunities exist with the last two options, to enhance the system, which may be worth considering. The greatest opportunity may be to cure a long term problem inherent in the system from the early days of operation. Other opportunities might be:

1. To upgrade design or material of components to reduce wear or unreliability,
2. To increase system capacity or output,
3. To improve environmental performance, etc.

Thus the final proposal will address a range of needs to allow the plant or system to operate for an extended lifetime and include some options and additional benefits.

An estimate of costs would allow some initial evaluation of corporate options.

After this proposal has been reviewed from the corporate perspective and declared the preferable option, the shutdown for refurbishment can be planned. This could be either a major front end shutdown or a number of annual projects combined with the routine maintenance shutdowns.

As can be seen, the assessment of remaining life must be done in a complete and sufficient manner if the total life extension process is to be reliable and economically useful to an organization. Conversely, to not consider such an opportunity to fully use available plant life (and thus avoid the costs of premature replacement) may be a very costly option.

The growing amount of expertise within the consulting companies, in the fields of component remaining life prediction, is making the potential for continued operation of components and systems in the future much greater.

Life Extension is thus likely to become an even better mechanical option in the future.

4. CONCLUSIONS

1. There is an increasing interest in the corporate benefits of Plant Life Extension.
2. The assessment of Remaining Life of components is the most important stage of the Life Extension Process.
3. Using the EPRI* philosophy, the process of confirming the necessary amount of remaining life can be done in the most cost effective manner.
4. Much of the remaining life assessment work can be completed in-house, until these resources are exhausted.

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ROLE OF MECHANICS IN STRUCTURAL INTEGRITY AND CORROSION ISSUES AFFECTING LWR PLANT LIFE EXTENSION: A TECHNICAL PERSPECTIVE

Y. S. Garud

S. Levy Incorporated
3425 S. Bascom Avenue
Campbell, CA 95008, U.S.A.

ABSTRACT

The mechanics of materials and structures plays an important role in majority of life prediction and structural integrity evaluations. In light water reactor systems this role gets integrated with effects of service environment and material aging; aqueous corrosion and irradiation are examples of such effects which require an interdisciplinary approach. Degradation mechanisms related to these effects have been identified as potentially significant to plant life extension (PLEX). In-service inspection and component monitoring are some of the cost effective steps emphasized in the life extension works; usefulness of the results of these steps can be enhanced through the methods needed for the component condition assessment. These methods, therefore, demand an ever increasing accuracy and improvements requiring a better appreciation and integration of the mechanics.

The above aspects of PLEX are critically examined in this paper with a review of key degradation mechanisms elucidating the role of mechanics. Current design basis and recent developments in the life prediction techniques are discussed identifying areas of further work. Stress corrosion cracking, corrosion fatigue, pressurized thermal shock, and long term material stability under thermal and irradiation aging are some of the phenomena emphasized in this work. The significance of mechanics in treating the sub-critical crack growth due to these effects is discussed. Also, from the point of view of final failure step, the role of impact energy and fracture toughness is examined with consideration of the time dependent changes in these properties. The additional objective of this presentation is to provide a perspective on the assessment of remaining life and safety margins with particular reference to the needs and challenges of the plant life extension.

INTRODUCTION

This paper reviews the necessary and useful role of structural and solid mechanics in the assessment of structural integrity issues of particular importance to the longevity of light water reactor (LWR) power plants—beyond that in the original design/analysis methods and requirements. The emphasis of this review is to provide a critical examination and an overall technical perspective of the key issues and related mechanics.

Most of these issues involve synergistic effect of thermo-mechanical loads and some environmental factor, like irradiation and corrosion, with the associated time dependent degradation.

First, the definition of plant life extension (PLEX) is made clear with reference to key events in the life of an LWR plant. The role of mechanics in various stages of PLEX is summarized. This is followed by a brief discussion of the key mechanisms of degradation and their inter-relation to mechanics, with PLEX im-

plications; these include irradiation and thermal aging, stress corrosion cracking, and fatigue. The usefulness and need for integration of mechanics are further elucidated with examples of structural integrity issues; pressurized thermal shock, cladding effects, thermal stratification, mitigation of stress corrosion, and treatment of dynamic loads are some of the examples covered. Finally, related areas of improvement and future work are briefly summarized.

The natural emphasis of this work is on the initiation and growth of sub-critical flaws, flaw tolerance and evaluation, and related safety margins. The work is limited to components and structures of metallic materials. The aging and mechanics of electricals (cables, equipment, etc.) and concrete structures (containments, supports, etc.), some of which are significant to PLEX work, are not covered in this paper.

WHAT'S PLANT LIFE EXTENSION?

A simple, non-technical view of plant life extension is essential to put in perspective the thrust, need, and goals for the related research work. Figure 1, meant to be mostly self-explanatory,¹ illustrates the key events, terminology, and stages in the time frame of the life of a power plant; the actual life of a specific plant will lie somewhere within the shown time frame.

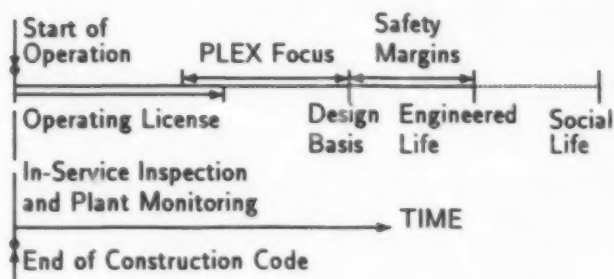


Figure 1 – Main events and terms related to LWR power plant life extension.

¹The term 'social life' refers to the period of continued acceptability of nuclear power industry as a whole and includes individual plant specific regions as well. This has an added significance after the Three Mile Island (TMI-2) and Chernobyl events of the last decade, and the Mihama incidence (in Japan) of last year. That is, the need to *successfully operate all plants, by doing everything possible on all fronts to avoid such instances (no matter how incredible they may appear on paper), has grown ever more.*

It is important to note that each of the events (in Figure 1) is determined by more or less an independent (different) set of rules and circumstances. It follows that the respective life spans are flexible and can be adjusted or impacted by changes in the rules and circumstances. The 40-year term of initial operating license (in the U.S.), granted under the *Atomic Energy Act of 1954*, was based in part on the (projected) period for investment recovery—most likely backed by some engineering analysis and judgment for the minimum safe life of a plant. Now, the actual service experience from various power plants over the past several years forms a solid data base providing for an improved analysis, better judgment, and identification of issues likely to affect the safety and reliability of future operations.

Therefore, PLEX may be defined as an *evolutionary process* involving technical, regulatory, and engineering activities related to plant monitoring, in-service inspection (ISI), technical assessment, and (remaining) life prediction methodologies, with a *common goal* of maximizing the economic plant life while maintaining adequate safety margins.

MECHANICS CONSIDERATIONS

The studies showing economic benefits and technical feasibility of longer term operation of existing power plants (EPRI, 1987 and Byron, 1990), although essential for consideration of the license renewal activity, constitute only the first steps of PLEX. Likewise, the prioritization of technical issues (Shah and MacDonald, 1989) and the development of PLEX related plant aging management programs (ASME, 1991) are needed in the early stages. In these, the role of mechanics is somewhat indirect and limited to providing the qualitative understanding of key issues and identifying significant factors for control or quantitative evaluation of the issues.

ISI and monitoring (of key components) are some of the cost effective steps emphasized in the recent PLEX works. Usefulness of the results of these steps is dependent on (a) the integrated understanding of mechanics aspects reflected in them and (b) the methods—either modifications of existing ones or new—necessary for the quantitative assessment of component condition. For example, the simplifications and uncertainties inherent in the analysis methodologies—for predicting the deformations, sub-critical cracking under environmental influence, and margins to final failures,

all of which require significant use of mechanics—should not be so overwhelming as to render the good ISI data less valuable, if not practically meaningless. Thus, improvements in the accuracy and predictive capabilities of the mechanics based, quantitative models and analytical tools need to be pursued to complement and enhance the utility of other PLEX efforts.

The formation and growth characteristics of flaws, especially in the components of (and within) the primary pressure boundary, under the combined influence of service conditions, are central to the PLEX work. Figure 2 illustrates a useful classification of flaws for the purpose of following discussion.

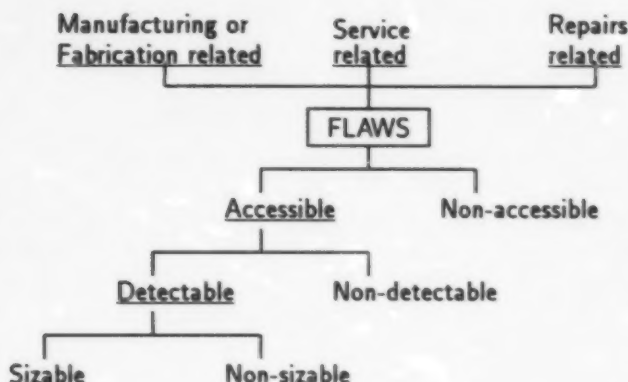


Figure 2 – Classification of flaws for condition monitoring and assessment.

It shows that the usual fracture mechanics type evaluations (e.g., Section XI of the ASME Code) really deal with the accessible, detectable and sizable flaws; for the remaining class of flaws one could, in principle, use a similar evaluation with hypothetical flaws (which introduces large uncertainties or unwarranted overconservatism). It is these areas and related potential for flaws which need alternate and careful considerations as well; protection of safety margins in such areas is also critical for longer service times. To a large extent, these margins are based on the *defence-in-depth* philosophy and indirect condition monitoring, dependent on such considerations as the *leak-before-break* concept. Evaluation of these characteristics of flaws and related safety margins are influenced by, and need direct input from, the mechanics of materials and structures—loads, loading rates, moments, deformation and stress-strain response, fracture behavior, etc.

SIGNIFICANT AGING MECHANISMS AND RELATED MECHANICS

A recent survey of Industry Reports (Byron, 1990) shows that, with only a few exceptions, the potentially significant age- and service-related degradation mechanisms involve some form of corrosion, cyclic loads (fatigue), and material stability under thermal and irradiation exposures. All of these are intimately related to the mechanics of materials, structures, and fracture evaluations as discussed in the following.

Irradiation Aging

Some of the pressure boundary components and support structures of major significance to PLEX of LWRs are continuously exposed to radiation, including fast neutrons (> 1 MeV) and gamma rays, known to induce changes at atomic and microscopic levels. Although these changes may not directly have any influence on the global quantities such as the applied or sustained loads and moments, their influence on local and internal material response must be properly taken into account in PLEX analysis.

Irradiation aging alters the local deformation and fracture properties mainly through its impact on point defects such as vacancies and solute atoms resulting in easier void nucleation and growth, altered dynamic strain aging characteristics, enhanced segregation at grain boundaries, etc. The yield strength, ultimate strength, and work-hardening rate are changed. In the case of austenitic steels the fracture strain is found to decrease even when the fracture is ductile. In ferritic steels the embrittlement is measured in terms of the brittle fracture characteristics such as the reduction in fracture toughness or (Charpy V-notch) upper shelf energy, and the increase in nil-ductility temperature.

Thermal Aging

The operating temperatures for critical components of LWRs are relatively low (around 250°C to 350°C) for most metals and alloys to exhibit dramatic property changes over considerable periods; however, 40 years of operation is about 35,000 hours which can, in some cases, begin to have measurable influence. In particular, cast stainless steels used in some piping components, main coolant pumps, control rod drive mechanisms, and reactor internals are suspected in this category of aging (Shah and MacDonald, 1989); also, the vessel cladding and some weldments are influenced.

Thermal aging refers to the possible changes in the deformation and fracture properties of materials due to the longer term thermal exposures; in metallic materials, these changes are mainly due to the microstructural effects of segregation (of minority elements of the alloys) and secondary (or newer) phase precipitations. The resulting thermal embrittlement is an important consideration in the evaluation of final stage of fracture. For example, the combined effect of irradiation and thermal aging on the toughness of stainless steel cladding is now expected to be more than that on the low-alloy vessel steels so that the analysis (e.g., initiation of ductile tearing) of such a composite must account for these property changes.

Fatigue

Flaw initiation and growth due to increasing number of load cycles is a well recognized degradation mechanism, called fatigue, in metallic components of which there are many with PLEX implications. The accurate assessment of fatigue in all its forms—low cycle fatigue, high cycle fatigue, thermal fatigue, corrosion fatigue—is critical to the determination of expected remaining life and safety margins. At present, this assessment is based on totally empirical models requiring large and relevant data base often translated into a single life curve assumed to be an upper bound.

With aging related changes and with effects like corrosion the need for updating the data base and the corresponding models continues to be critical. In addition, over the past several years of service, a number of unanticipated loading events (not factored in the original design calculations) have been identified which have the potential for recurrence. These events include, for example, fast thermal cycles due to flow stratification and thermal striping in feedwater systems, flow-induced vibrations, and thermal streaming on the inside of a vessel wall near inlet nozzles. Analysis of peak local stresses and determination of the cyclic stress spectrum for these events and their accounting in the fatigue usage are areas important to PLEX.

Corrosion

Corrosion, in general, and corrosion fatigue (CF) and stress corrosion cracking (SCC) in particular, are the most prevalent aging mechanisms of concern in LWRs

due to the aqueous environments. SCC refers to the conjoint action of steady or slowly applied loads and corrosion reactions so as to produce accelerated cracking (relative to that produced, if at all, if these factors were acting alone); CF, although likely to be mechanistically same as SCC, refers to enhanced fatigue kinetics under corrosion and cyclic loads. These mechanisms may or may not include contributions from the effects of hydrogen on local deformation and fracture response. Assessment of flaw characteristics due to SCC and CF requires significant inputs, such as the applied and residual stresses, strains, and strain rates, from mechanics analysis.

Here, thermal aging can have an additional impact on the initiation and sub-critical growth of cracks because of the resulting low temperature sensitization (LTS). LTS, which increases the susceptibility to stress corrosion and corrosion fatigue, is known to occur in certain austenitic stainless steels and nickel based alloys. Synergistic effects of thermal aging and irradiation aging are likely to influence the susceptibility as well; these progressive changes are such that empirical approaches (e.g., Arrhenius or exponential models of aging) based on the assumption of one dominant mechanism may be inadequate, at best. Irradiation, by itself, has been known to affect the water chemistry and, especially, the electrochemical corrosion potential.

Recent reviews of SCC related data and field observations have confirmed that, at least in some material conditions, irradiation can cause further acceleration in flaw growth, or expand the load-environment combinations under which SCC can occur; this combined mechanism is known as irradiation assisted stress corrosion cracking (IASCC). The fact that IASCC does not require high degree of material sensitization or high tensile stresses clearly shows why one must consider the above effects in procedures for estimating component life and flaw tolerance.

Two other significant mechanisms of corrosion related damage are the erosion-corrosion and the microbiologically influenced corrosion (MIC) which lead to wall thinning and/or surface fouling. Here, although the main factors are related to the thermal and hydrodynamics (flow) conditions, the effect of (kinetics and extent of) degradation on structural integrity of susceptible components may need to be accounted for in some cases as well.

PLEX RELATED USE OF MECHANICS: SOME EXAMPLES

One of the most extensively examined issues is that of the flaw tolerance of a reactor vessel under pressurized thermal shock (PTS) conditions (USNRC, 1987); this issue continues to be investigated because of its high safety significance. PTS is an event under which the vessel wall is subject to a sudden over-cooling transient while the pressure is still high, thus resulting in large thermal and total stresses; under certain conditions these high stresses may be enough to potentially cause rapid crack growth (of a pre-existing flaw), especially when the wall material has undergone considerable irradiation embrittlement. Determination of the combined stresses, assessment of irradiation effects, and brittle fracture or ductile tearing resistance evaluations are important considerations; also, the respective gradients through the thick wall of vessels influence the analysis of possible crack arrest. The residual stresses (generated due to thermal property differences between the vessel cladding and vessel materials) and a through-the-cladding flaw also present a unique set of circumstances requiring careful and detail analysis of flaw growth. Results of such evaluations can be expected to influence major decisions including that of the (downgrading of) allowed pressure-temperature profiles for plant operations.

Thermal-structural analysis for the case of flow stratification, such as in the feed-water piping components and pressurizer connections (Shah and MacDonald, 1989), is an example where applications of mechanics have been developed to determine stress cycles and to compute the fatigue usage (Clayton and Irvine, 1987). In addition to this relatively local cyclic thermal stratification leading to high-cycle fatigue and crack growth, more global thermal stratification has been observed with resultant low-cycle fatigue, large displacements or high stresses (USNRC, 1991). In general, the resulting thermal stresses have large axial components due to the circumferential variation of temperatures normally not addressed by design procedures. Similarly, thermal streaming in the reactor vessel downcomer annulus flow produces nonuniform circumferential temperatures in the vessel wall. The potential for streaming in LWRs has recently received greater attention due to the implications for circumferential flaw growth under PTS as well; this phenomenon has been previously recognized and analyzed in other reactor designs.

Stress corrosion cracking of pipe welds, heat exchanger tubes, nozzles, and reactor internals is a dominant generic issue compromising the integrity by continued effect on the initiation and sub-critical growth of flaws; the kinetics of this mechanism has never been explicitly factored into the design or analysis. Although tensile stresses are considered necessary for SCC the extensive work on this issue has shown that the key mechanics parameter is the local rate of deformation which is needed for a rational assessment of SCC. This has led to some strain-rate based quantitative models which continue to be refined (Ford, 1987 and Garud, 1991a). Integration of the mechanics of time dependent visco-plasticity to define the strain-rate response of interest is important to this issue (Garud, 1991a).

The mechanics based remedial or mitigation measures have already been developed and implemented in some cases to improve the component life which would otherwise be shortened by SCC. For example, in the case of heat exchanger tubes the technique of shot-peening to induce compressive stresses has been claimed to be beneficial and, in the case of pipe welds, the so-called mechanical stress improvement process, MSIP, and induction heating stress improvement, IHSI, have been considered as cost-effective measures. Use of these methods for already flawed components requires more careful mechanics evaluations.

One area of the original design procedures where large conservatism, with some counterproductive effects, has been identified is that of the treatment of seismic and reversed dynamic loads; the related experimental and analytical investigations have led to some significant changes in the evaluation of this category of loads (Guzy, 1988 and Tagart, 1990). Fatigue with ratcheting, as opposed to plastic collapse, has been determined to be the relevant mode of failure. More explicit and quantitative determination of ratchet strains (Garud, 1991b) and related impact on the fatigue life are some of the useful areas of application of mechanics. Simplified methods of analysis are being developed taking into account the effects of structural and material non-linearity and inelasticity (Jaquay and Castle, 1990 and Seshadri, 1991).

Some New or Unresolved Issues

Perhaps the most difficult part of assessing the long-term aging effects is to know exactly what the current state of material is and to estimate what it would be at the projected end-of-life; this necessitates the use of some accelerated test conditions and the correspond-

ing interpretation of results. The problem is made more complex by the possible synergistic effects between loading conditions (stress states), thermal cycles, and irradiation exposures which are all but impossible to simulate.

For example, in relation to SCC, fast fracture, and crack arrest phenomena known to be influenced by the local deformation characteristics, the effects of dynamic strain aging have received very little attention. Also lacking is any quantitative evaluation of the extent to which stresses and strains under operating conditions affect the irradiation aging and thermal aging (by providing additional driving force for the underlying physical mechanisms). Further, effects of the more prevalent multiaxiality of stresses and rates of loading on irradiation embrittlement need to be examined.

One of the key effects, known as the low fluence, low temperature (LFLT) embrittlement, has been recently identified showing greater than expected irradiation damage where time dependence, neutron spectrum and fluence-rate may need more consideration (Cheverton et al., 1989 and Lucas et al., 1991).

In the current flaw evaluation procedures, one major simplification that needs to be re-evaluated is that of the assumed *self-similar* growth of a crack. From the preceding discussion it is clear that conditions exist, especially for thermal transients, where flaw growth *along the surface* may be much different than that in the through-wall direction. This can be important in the analysis of remaining life, leak versus break considerations, and safety margins. Also, the relevance and accuracy of using long crack (fracture mechanics type specimen) data to analyze kinetics of short crack growth or to analyze cases involving significant biaxial loading and mixed mode cracking have raised some questions which need critical mechanics considerations. Local constraint effects and crack arrest considerations have implications, not yet fully resolved, in the application of data from laboratory specimen geometries to the actual component geometries. Many crack-tip parameters (e.g., fracture initiation toughness), notwithstanding the constitutive response assumed to prevail locally, are treated as if they were *material properties*, independent of the global loading type, the component geometry, and local constraints.

One aspect of the stress corrosion cracking which deserves more attention is the assessment of multiple crack initiations and the possibility of their subsequent coalescence (Parkins and Singh, 1990) leading to sudden increase in the flaw size. Due to their initial small

size and extremely slow growth rate over very long periods these flaws may not become apparent in the normal course. The stress-strain and deformation fields associated with a network of short cracks, and conditions or factors affecting their coalescence are areas where contributions from the mechanics can be useful.

DISCUSSION

Many phenomena (or degradation mechanisms) have been observed during the first 20 or so years of LWR plant life which were not explicitly addressed (or accounted for) either in the construction/design code or in component analysis. Most of these involve (a) synergistic effects of environmental and loading factors as in SCC and corrosion fatigue, or with material changes as in IASCC and irradiation embrittlement, or (b) unanticipated damaging load cycles as in thermal striping, PTS, and fluid-structure interactions. These aspects of past experience provide enough cause for concern that similar unanticipated failure mechanisms or, in the case of now anticipated mechanisms, misjudged kinetics or rate effects may be observed in the future long-term service, perhaps with greater influence of time-dependent aging factors.

On the other hand, the same past experience has led to much better understanding of the degradation mechanisms and has resulted in some effective mitigation actions, alternate methods of analysis, and the possible availability of useful data bases; in some cases, areas of excessive conservatism in designs have been identified as well. This provides for a good basis essential for meeting the PLEX goals. Clearly, in all of these efforts the mechanics of materials and structures play an important role, but not in isolation.

The uncertainty band of extrapolation (based on empirical models) and the reduced safety margins (due to aging) should not be allowed to overlap. The uncertainty can be reduced by (1) more accurate and realistic analysis of actual loads, cycles, stresses, etc., (2) development and integration of relevant, useful, quality data bases, and (3) mechanistic (physically based) interpretation of such data and its integration in the quantitative models. The safety margins can be increased by (1) mitigation and remedial measures (shot-peening, water chemistry changes, material changes through annealing, etc.), (2) adjustments to operating characteristics of pressure, temperature, and start-up/shutdown cycles, and (3) improved ISI.

In general, much has been learned about the environmental (water chemistry) conditions and loadings (thermal striping, fluid-structure interactions, etc.)—with correspondingly improved monitoring and assessment techniques, but not as much or enough about the material aging and time-dependent phenomena. More work is needed in this area to enable accurate and confident extrapolation (in time) of key properties influencing the sub-critical and critical flaw growth characteristics. It seems important that the PLEX goals will be better supported by (a) more explicit consideration of local (e.g., near-crack-tip) stress-strain-deformation and fracture response, e.g., through a damage mechanics approach and (b) more explicit integration of mechanistic (physical) understanding of the underlying phenomena.

CONCLUDING REMARKS

Past LWR experience has shown that most of the major integrity issues can be attributed to the actions of unanticipated (or not explicitly addressed in the original design/analysis) loading events and environmental (corrosion) effects; certain material changes due to thermal and irradiation aging actions are also becoming more apparent. Better understanding and integration of these factors with the application of mechanics, especially in the quantitative assessment of flaws, are essential for the success of PLEX. The above summary was an effort to elucidate the need, scope, and areas of improvements where material and structural mechanics can be expected to play an important role in meeting the PLEX goals.

More specifically, mechanics considerations need to be utilized in the following: (a) identification of damage mechanism(s) and associated critical regions, (b) identification and determination of key engineering factors and their integration in methods for determining rates (kinetics) of degradation, (c) development of the underlying physical basis and related parameters, (d) integration into mechanistically based models of damage assessment, (e) determination of alternate operational characteristics (of pressure-temperature transients and power levels), (f) identification of excess conservatism and alternate methods to relax some requirements, (g) development of more accurate methods of linear, non-linear, static and dynamic analysis, cyclic stress analysis, time dependent analysis, local and crack-tip analysis.

Non-linear analysis and inelastic (visco-plastic) response evaluations were not specifically and explicitly noted only to emphasize that these are no longer optional and will continue to be a growing and significant part in applications of the mechanics. Also, probabilistic evaluations need inputs from mechanics, especially in the load-resistance interference type analysis—as opposed to the frequency or rate (per reactor year of operation) type analysis—which can provide more absolute valuations of conditional probabilities and safety margins in a specific application.

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LIFE EXTENSION AND CONDITION ASSESSMENT
OF UTILITY STEAM GENERATORS

M. Hammoud, P. Eng.
Regional Service Engineer
Babcock & Wilcox
581 Coronation Blvd.
Cambridge, Ontario

ABSTRACT

Many utilities are increasingly becoming interested in prolonging the lives of their steam generating equipment beyond their expected operating lives. Babcock & Wilcox offers different approaches to life extension and condition assessment programs all of which are designed to suit customer needs. This paper discusses the various approaches including a typical phased approach program. Some of the standard and patented B&W condition evaluation procedures are presented.

INTRODUCTION

Recently, many electric utilities are opting to extend the life of steam generating equipment built in the 50's, 60's and 70's by 15 to 20 years due to the increased cost of new construction and increased demand for electrical power. Other purposes of a life extension program is to minimize forced outages and develop action plan for component replacements and modifications to meet current standard engineering designs.

- Equipment testing (performance assurance program).
- Evaluation, analysis and recommendations.
- Engineering studies and life extension action plan.
- Implementing upgrades, alterations, and recommended operating procedures.

Few utilities are implementing this wide approach.

DIFFERENT APPROACHES

There are different approaches to perform a life extension and condition assessment studies. It is dependent on the owners needs and future plans.

A single component approach looks at high priority and critical components. It is difficult to measure the return in benefit from this approach.

A single unit approach looks at high priority units first, such as critical units with major problems or cycling units. The return in benefit using this approach can be assessed.

The most effective approach is the system wide approach where a service life extension program normally consists of six (6) stages.

- Evaluation and planning
- Outage inspection (boiler fitness survey).

TYPICAL PHASED APPROACH PROGRAM

A typical boiler fitness survey normally consists of three phases:

In phase I, pre-outage period, an outage inspection and testing plan will be developed. The plan will be based on reviewing the boiler maintenance and operating history, problem areas identified by the station staff, and on past experience of problem areas and inspection procedures.

In phase II, outage period, all site work and inspection will be performed according to the plan drafted in phase I.

In phase III, post outage period, the data and information collected in phase II will be analyzed and processed. A formal boiler evaluation and condition assessment report will be issued with recommendations.

STANDARD CONDITION EVALUATION PROCEDURES

The life expectancy of a utility boiler above 1200 psi is less than 45 years. Common failure mechanisms for low temperature (<900°F) components are corrosion/erosion and fatigue. For higher temperature components (>900°F), common failure mechanisms are creep, fatigue, embrittlement, corrosion/erosion and overheating. Both destructive and non-destructive examinations are used to assess the condition and remaining life of a steam generator. Some of the established industry standard Babcock & Wilcox examinations included in a life extension and condition assessment survey are:

- Dye penetrant and magnetic particle tests of surfaces and welds in headers, piping and tubes to check for any cracking or discontinuities in the material.
- Ultrasonic testing to check wall thickness of tubes and piping to detect any wastage due to erosion or corrosion. Ultrasonic testing is also used to detect any cracking or flaws in weld or base material.
- Remote video probe inspection of internal surfaces of critical headers and tubes. Any internal deposits, scale, flaws or cracks in the material can be detected using this method.
- Replication of a component surface microstructure by taking a "finger print" image on a plastic film. The "replicas" are examined in a laboratory using standard optical or electron microscopy to determine if there are any microstructural changes due to a failure mechanism such as creep, creep fatigue and thermal fatigue. The principle of the surface replication technique is illustrated in Figure 1.

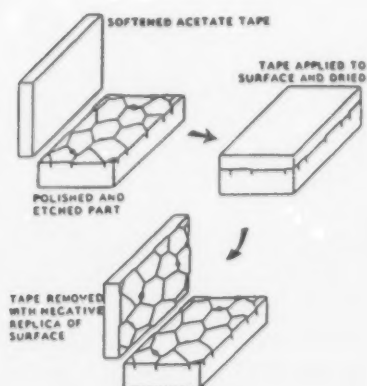


Figure 1
Replication Technique

Creep damage is usually observed in high temperature, high stress components. It typically progresses in stages with increasing strain (plastic deformation) over the life time of the component. Material damage manifests itself in the micro structure in the form of voids or cracks. Voids may form after mid-life, which then subsequently link up to form cracks. The replication technique is instrumental in detecting the voiding in the microstructure and subsequent void link up and formation of cracks.

The first published attempt to relate creep life consumption of plant components to cavitation was that of Neubauer and Wedel.* They characterized cavity evolution in steels at four stages, i.e. isolated cavities, oriented cavities, linked cavities (microcracks) and macrocracks. They estimated the approximate time intervals required for the damage to evolve from one stage to the next under typical plant conditions. They formulated recommendations corresponding to the four stages of cavitation. Creep-life assessment based on cavity classification is illustrated in Figure 2.

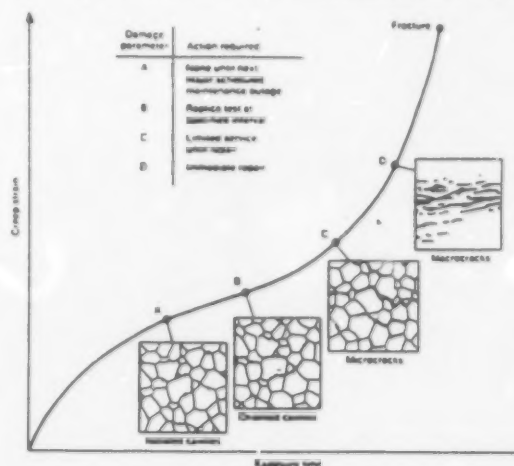
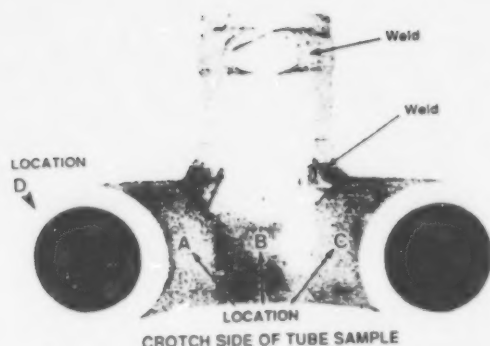


FIGURE 2

Assessment of Creep Damage

- Tube samples are usually removed from critical and known problem areas in the boiler. Usually from locations of highest temperatures, dissimilar metal welds and erosion/corrosion areas. Tubes are subjected to a series of laboratory metallurgical examinations to determine its general condition, remaining life and microstructural condition. Results of an optical metallography test of a transverse section of a tube sample is shown in Figure 3.

* B. Neubauer and U. Wedel, "Restlife Estimation of Creeping Components by Means of Replicas," *Advances in Life Prediction Methods*, American Society of Mechanical Engineers, New York, 1983, pp. 307-314.



BIFURCATE SAMPLE BS3
SECTION 3B



Figure 3

Cracks In Heat Affected Zone of Weld

PATENTED B&W CONDITION EVALUATION PROCEDURES

The following outlines some unique and patented B&W examinations usually included in a life extension and condition assessment survey.

NOTIS: (Non-destructive Oxide Thickness Inspection System)

High temperature (above 900°F) and pressure tubes normally fail due to creep rupture possibly accelerated by the effects of corrosion and internal buildup of oxide scale on the steam side of the tube. Creep rupture is a time dependent failure mechanism. The tubing is evaluated with a sophisticated, microprocessor-controlled ultrasonic test (Fig. 4).

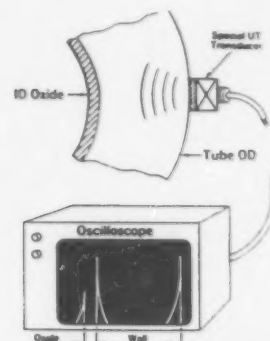


Figure 4

Microprocessor Controlled Ultrasonic Equipment

This test not only measures the thickness of the tube metal, but also the thickness of the steamside oxide scale. The oxide scale allows the tubes to experience higher temperatures which reduce its operating life by accelerating the creep-rupture failure mechanisms.

The data collected during the test are converted, utilizing known operating and design conditions, into remaining life estimates by use of a specially designed computer program (Fig. 5). The effects of steamside oxide growth rate are taken into account. The increase in temperature with time is roughly proportional to oxide thickness. Wall thinning due to material wastage is also taken into account since it will elevate the stresses on the tube.

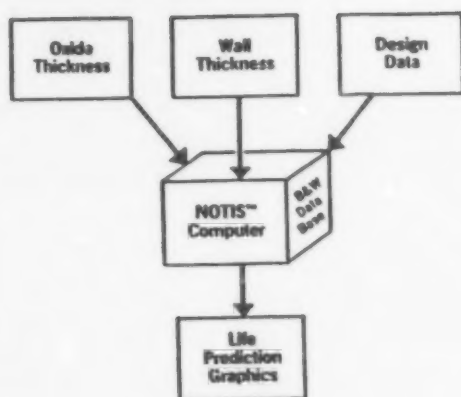


Figure 6

Prediction of Remaining Life

FHYNES (Furnace Hydrogen Damage Nondestructive Examination System)

Hydrogen damage begins on the inside surface of furnace tubes subjected to the high temperature furnace gases. It is usually associated with under deposit corrosion. Hydrogen produced in the corrosive reaction between the fluid and tube metal diffuses into the underlying metal, decarburizing the tube from the inside and also causing microcracks in the tube metal.

FHYNES is a fast and non-destructive examination which employs the shear wave ultrasonic method to detect changes in sound transmission properties associated with the microcracking found in hydrogen damaged tubes (Fig. 6)

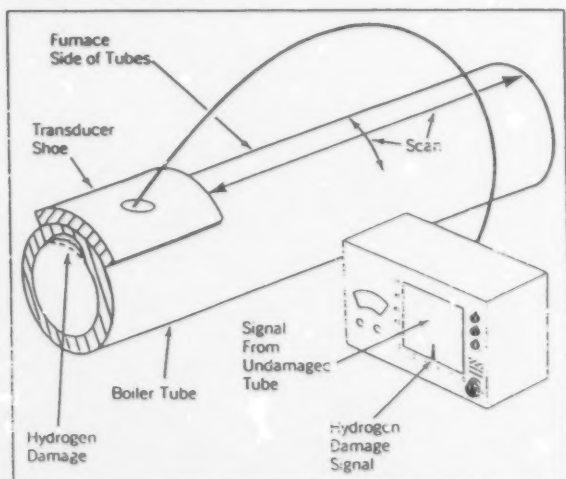


Figure 6

Shear Wave Technique is Highly Sensitive
For Detecting Hydrogen Damage

HONE AND GLOW

This method is used to evaluate the condition of high temperature headers. It is a fluorescent dye-penetrant examination of tube bore holes which provides a mean to visually locate internal ligament cracking using a fiber optic probe (Fig. 7)

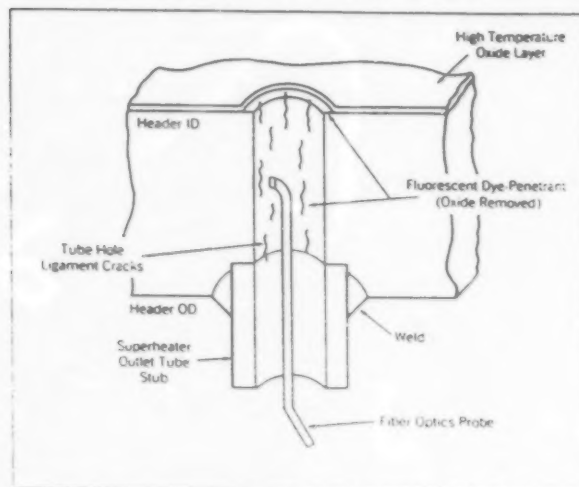


Figure 7

Diagram of Tube Bore Hole Exam With High Temperature Oxide Layer Removed

The above are only a few of many patented techniques which B&W has developed recently to enable its customers to make timely, better informed, cost-effective decisions concerning the condition and maintenance of their steam generating units. Other "high technology" services offered by B&W are:

- MANTIS - Corrosion/Erosion Mapping
- FURNIS - Furnace Tube Relative Deposit Loading
- Deep Eddy - Tubular Airheater Wall Thinning Test
- Acoustic Ranger - Tubular Airheater Leak Locator
- Crack Sizing Technology and Crack Growth Analysis for Headers and Piping
- Cycling Analysis
- Upgraded Attemperators/Headers

SUMMARY

The main objectives of a B&W life extension and condition assessment program are to:

- Determine the fitness of equipment for a continued and reliable service by investigating the various causes or failure and damage.
- Recommend modifications to existing equipment, and to operating and maintenance procedures, which will lead to reliable and extended service life.

**LIFE ASSESSMENT OF TRANSALTA UTILITIES'
WABAMUN GENERATING STATION - A STATUS REPORT**

By

B.D. Holter, TransAlta Utilities Corporation
J.S. Kastner, Monenco Inc.
D. Scoulding, Monenco Inc.

This paper presents the status of the life assessment study of TransAlta Utilities Wabamun Generating Station. The plant is a pulverized coal fired thermal plant consisting of 2-66 MWe, 1-150 MWe and 1-300 MWe units commissioned in 1956, 1958, 1962 and 1968, respectively. The primary function of the assessment is to evaluate the viability and economics of operating the existing 4 unit plant at base load initially to the year 2003 (the planned retirement date) and then in subsequent 5 year increments to the year 2018.

The 3 phase approach includes; Phase I: a compilation and review of all available data to assess both the viability and upgrades necessary for operating all of the plant units until the year 2003, Phase II: an extension of the Phase I results to identify the requirements, methods, techniques, time and costs to realize each of three five year post 2003 life extension periods, and Phase III: investigation of the cost effectiveness of each of the five year post 2003 life extension increments.

The study does not take into account any new environmental control systems or modifications which either are, or may be, required in order to operate the plant to 2003 or beyond.

PLANT DESCRIPTION

The Wabamun Generating Station was the first pulverized coal fired boiler/condensing steam turbine thermal generating station built by TransAlta Utilities (then Calgary Power Limited). It represented a major departure for what was, at that time, a primarily hydro power based utility. The basic plant operating data and history are summarized in Table 1.

The plant is located on the north shore of Lake Wabamun, approximately 60 km west of Edmonton, Alberta, and uses the lake water directly for condenser cooling. The development of the Wabamun plant required the concurrent development of the adjacent Whitewood coal mine which uses strip mining techniques to mine subbituminous coal having a typical average ultimate analysis of that shown in Table 2.

TABLE 1
Wabamun Generating Station Data Summary

Item		Unit 1	Unit 2	Unit 3	Unit 4
Year Commissioned		1958	1956	1962	1968
Capacity	MWe	66	66	150	300
Fuel		Natural gas and coal ⁽¹⁾	Natural gas and coal	Coal	Coal
Boiler Manufacturer		Babcock & Wilcox	Babcock & Wilcox	Combustion Engineering	Combustion Engineering
Turbine-Generator Manufacturer		Metropolitan-Vickers	Metropolitan-Vickers	AEI	AEI
Main Steam Conditions					
- pressure	kPa (gauge)	5860	6200	13100	16550
- temperature	°C	500	500	538	538
Hot Reheat Steam Conditions					
- temperature	°C	N/A	N/A	538	538
Condenser Backpressure	in Hg	3	3	3	3
Approximate Operating Hours to Date	Hours	200000	247000	206000	169000

Note: (1) Coal firing added in 1983
(2) N/A = not applicable

TABLE 2

Whitewood Coal Ultimate Analysis

Moisture	17.64 - 21.75
Ash	12.39 - 16.16
Carbon	47.65 - 51.83
Hydrogen	4.10 - 5.09
Nitrogen	0.58 - 0.64
Sulphur	0.17 - 0.29
Oxygen	10.61 - 12.15

PLANT OPERATION

The Wabamun units have all operated continuously since commissioning. They were initially operated on base load until the mid to late 1970's, when the newer Sundance and subsequent Keele units placed the Wabamun units onto a reduced capacity status until the late 1980's. At this time a combination of increased electrical power demand and lower Whitewood Mine coal costs brought the Wabamun units back onto base load status. These units presently have a five year average availability factor of over 85%.

LIFE ASSESSMENT/RETIREMENT DATE

TransAlta has determined that, at the present coal consumption rate, the currently leased Whitewood mine resources will expire in the year 2003, although adjacent coal resources have been identified which would permit operation past the year 2003. In addition, TransAlta has in the past six to eight years investigated both the future need for the Wabamun Plant and its approximate remaining life. These investigations resulted in remaining life estimates of five to ten years (i.e. until 1990 to 1995) for some major plant components, subject to the results of suggested subsequent investigations.

This combination of coal lease duration and existing life expectancy studies has prompted TransAlta to investigate several alternatives for continuing operations at the Wabamun plant site up to and beyond the year 2003. The current study investigates continuing with the present conventional pulverized coal firing configuration to initially achieve the year 2003 and then extending the operating life in three-five year increments until the year 2018.

STUDY REQUIREMENTS

The following study assumptions and requirements were established prior to the initiation of Phase I of the study; a) No compromising of employee or plant safety; b) Expected condition of plant in 2003 to be based upon expected maintenance and replacement work to be done prior to 2003; c) Plant to maintain existing reliability; d) Plant to be run on base load operation; e) Coal will be available; f) No changes to cooling water source and system; and g) No changes to existing environmental requirements for plant.

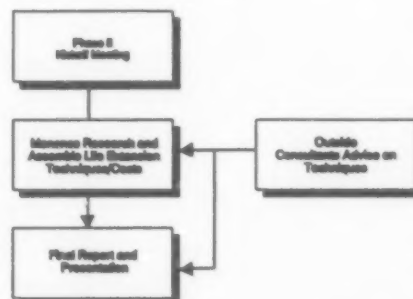
The study report groups the plant units into Units one & two as similar units; Unit three; and Unit four as separate units. For each unit/group, the following information is to be provided: a) Methods of life extension applicable; b) Cost of life extension methods; and c) Lead time of life extension methods. The study is to assess the requirements and cost to operate each unit beyond the year 2003 by 5 years; 10 years; and 15 years.

STUDY PHASES

Phase I required Monenco to research and compile TransAlta's available existing data on the equipment status for all four units at the Wabamun Plant. These data include all tests, reports and studies performed to date relating to the evaluation of remaining plant life; the existing or future planned equipment or parts replacement up to the year 2003; past inspection and maintenance; and proposed future (major) maintenance. The sequence of execution of Phase I is shown on Figure 1.

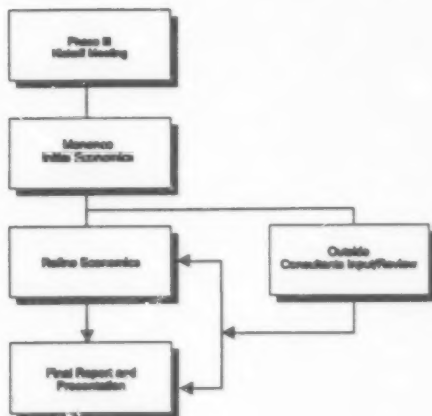
Figure 1
Wabamun Life Extension Study - Phase I

With the acceptance of Phase I, Phase II will identify the methods, techniques, time and costs required to continue operation to 2003 and the similar requirements for attaining the three post 2003 life extension periods specified. No technical or economic evaluations will be carried out during this Phase. The sequence of execution is shown on Figure 2.

Figure 2
Wabamun Life Extension Study - Phase II

Based on satisfactory Phase II results, Phase III will evaluate the life extension alternatives presented in Phase II to determine which, if any, are cost effective in the pre 2003 period for each of the post 2003 life extension periods specified. The sequence of execution is shown on Figure 3.

Figure 3
Wabamun Life Extension Study - Phase III



EXECUTION

To date, Phase I has been completed and the draft report is under review by TransAlta. This work was carried out by reviewing existing TransAlta operations files and records of all the major components and systems identified below and by conducting, in association with an outside consultant, a site inspection of the boiler and steam piping related systems.

The data was categorized into the following five major areas: boiler and boiler related auxiliaries; steam turbine; steam piping; generator and major electrical equipment.

The boiler area was generally bounded by the feedwater inlet connection, the coal feeder inlets, the air preheater outlet(s), the superheater outlet terminal points, the reheater inlet and outlet terminal points (where applicable), and the ash discharge points. The steam turbine area included only the steam turbine itself and not the generator or condenser.

The steam piping area included only the steam piping interconnecting the boilers and steam turbines.

The generator area included the station, rotor and cooling system.

The major electrical equipment area included the automatic voltage regulator (AVR) and excitation systems, 4.16/2.3 kV switchgear and motor starters, 480 V switchgear, and 480 V motor control centres.

Monenco engaged two outside consultants to assist in the review and evaluation. The boiler and boiler related auxiliaries area and the steam piping area were reviewed by Mr. David Winship of "Combustion and Thermal Power Consulting Engineering" and the steam turbine area was reviewed by Mr. Frank Quinn of "Steam Path Services". The balance of the major areas were reviewed by Monenco personnel. This approach provided TransAlta the advantage of using recognized specialists together with Monenco's intimate "original plant designer" knowledge of the plant.

RESULTS OF PHASE I

The results of Phase I can be summarized as follows:

- Boilers and auxiliaries: all can be made to operate until 2003 with the existing maintenance program;
- Steam turbines: all can be made to operate until 2003 with the existing maintenance program;
- Steam piping: some potential problems, further investigation is required;
- Generator: no major problem areas identified;
- Major electrical equipment: existing equipment can probably be operated until 2003 but not significantly longer.

More specific comments for each unit are provided in the following sections.

Units No. 1 and No. 2

The following major components were recommended for inclusion in the life extension/maintenance inspection program for the boiler area:

- steam and mud drums;
- superheater and desuperheaters;
- superheater support and alignment spacers; and
- steam drum and superheater headers safety valve supports.

Significant items that were noted during a visual inspection of the boiler while in service and a review of operating and maintenance records were:

- Ash build-up on the leading superheater elements (bull nose) was noted to be between 10 and 20 inches deep. Continued surveillance of this location appears to be necessary as if this deposit remained in place for a long period of time, it could harden and be difficult to remove as well as limit the expansion of the superheater elements.
- The number of furnace wall soot blowers on units No. 1 and 2 are different and unit No. 1 has no wall blowers on the front wall. If furnace wall cleaning is determined to be unsatisfactory, additional blowers should be added or the location of existing wall blowers changed.
- Buckstay attachment failures have been noted. The extent of failures and repairs completed this year will be a factor in recommendations for life extension work.
- The casings and insulation are in relatively bad condition with some sections having been maintained or replaced. An Asbestos Management Project is currently under way to replace parts of these casings (at the present time being carried out on unit No. 2) and to remove asbestos insulation. The design of these units does not incorporate an inner skin casing and therefore significant leaking through the insulation could result in massive infiltration and ultimately a reduction in unit capability and efficiency. For the purpose of this study, it is anticipated that when the insulation and casing renewal is complete it will be suitable to last until the year 2003.
- No records of superheater supporting spacers or alignment spacer conditions have been found. The condition of these spacers should be monitored on the next annual outage to determine if any widespread replacement is required due to oxidation or corrosion damage.

An undesirably high number of economiser roof attachments failures have been reported on Unit No. 2. The detail design of the economiser roof is different between units Nos. 1 and 2. A program is underway to repair unit No. 2 at its next annual outage. It is expected that this repair will resolve this problem.

Assessment by both the outside consultant and the turbine manufacturer indicate that the unit can be operated as required until 2003 under the present maintenance scheme.

Bechtel had examined the main steam piping in Unit No. 2 during 1984-1985. At that time the unit had 201,000 hours of operation. Bechtel performed magnetic particle testing (MT) at 26 locations, ultrasonic testing (UT) at 26 locations, circumferential measurements at 24 locations, hardness tests at 24 locations and replications at 12 locations. Chemical analysis of piping material, forgings and castings was also determined. The Bechtel report suggested that the piping was likely suitable for an additional 5 years of operation (to mid 1990). They advised that several areas of the piping were in late second stage creep or early third stage creep but that no severe creep damage was noted. This will be investigated further in Phase II. No work was done on Unit No. 1, but as they are similar, it is expected that similar assessments apply.

Regular inspections and maintenance of the generators and their auxiliaries have been carried out on both units during annual turnarounds. A detailed inspection of Unit No. 1 was carried out in early 1990 and no major problems or damage found. Following a stator earth fault trip, the stator on Unit No. 2 was completely rewound in mid 1987. A major inspection of Unit No. 2 has recently been completed with no major problems or damage being found.

All major electrical equipment should be able to attain 2003 if adequate (used) spares can be obtained as new parts for any of the major equipment items are generally unavailable. This equipment includes: AVR & excitation systems, 416/2.3 kV switchgear, motor starters, 480 V switchgear, and 480 V motor control centres.

Unit No. 3

The following major boiler components were recommended for inclusion in the life extension/maintenance inspection program for the unit boiler areas:

- steam and mud drums;
- superheater and reheater desuperheaters;
- superheater and reheater support and alignment spacers; and
- drum, superheater and reheater headers safety valve supports.

Significant items that were noted during a visual inspection of the boiler while in service and a review of operating and maintenance records were:

- No records of superheater and reheater supporting spacers or alignment spacer conditions have been found. The condition of these spacers should be monitored on the next annual outage to determine if any widespread replacement is required due to oxidation or corrosion damage.
- When the hot electrostatic precipitators were installed on Unit No. 3, it was indicated in a report that the duct and hopper loadings due to fly ash accumulation had been substantially increased beyond the original design conditions and it was recommended that the support steel and hanger rods be reviewed for compliance to industry standards.
- Waterwall soot blower erosion has occurred. Combustion Engineering has developed a modified firing system referred to as concentric firing, which can reduce the water wall deposits in the burner and near burner region by up to 50%. This concentric firing system also reduces the generation of NO_x from nitrogen contained in the fuel. As this is a relatively low cost retrofit, it may be cost effective considering only the reduction in waterwall maintenance adjacent to the sootblower and associated forced outage costs.

- The primary superheater is subject to erosion damage due to a high average gas velocity through the surface. Throughout the life of the unit, attempts to minimise the erosion damage have been made by installing shelves and shields and a clean air test has been run on the unit to determine regions of high velocity. In the last decade, substantial advances have been made in the reduction of erosion damage due to localized concentration of ash and localized regions of high gas velocities. It is recommended that the original equipment supplier, who has substantial experience in this area, be requested to undertake a study to minimise erosion damage and to present a cost for carrying out the recommendations.

Assessment by both the outside consultant and the turbine manufacturer indicate that the steam turbine can be operated as required until 2003 under the present maintenance scheme.

Bechtel undertook non-destructive examination (NDE) of the main steam (MS), hot reheat (HR) and cold reheat (CR) piping in mid-1984. At that time the unit had 151,000 hours of service. Bechtel pointed out in their report that the piping had been designed under pre-1963 rules regarding allowable stresses in P11 material and that changes in 1963 had reduced allowable stresses by 30%. Bechtel examined 12 locations on the MS, 11 locations on the HR and 9 locations on the CR lines. They also performed accelerated creep testing on a sample removed from the piping. Two cracks were identified during the examination, both of which were removed easily by grinding. Ultrasonic examination showed no evidence of pipe thinning although one location with a wall thickness 17% under design specification was identified. This location (on the HR line) was highlighted for special attention in future NDE work. No further information was found in regard to this point.

Based on the metallography and accelerated creep testing, Bechtel suggested that the remaining life of the piping was 112,000 hours (i.e. 17 years (to 2001) at 75% operating factor). They recommended that welds not examined during their investigations be examined within 24 months, that all MS and HR castings be checked with UT, visual and MT examinations during the next turnaround and that a complete re-assessment of the piping be performed after ten years (i.e. 1995).

In 1986, Hanson Materials Engineering performed NDE at a total of 25 locations on castings for the MS, HR and CR lines for Unit #3. These investigations revealed numerous cracks on both the HR and CR systems. Cracks were removed by grinding on the HR lines and on the interceptor valves. Attempts to remove cracks on the CR castings were not successful and these castings were all subsequently replaced on this unit.

Regular inspections and maintenance of the generator area have been carried out during annual turnarounds. A major inspection was carried out in early 1991 with no evidence of major problems or drainage being found.

The findings in regard to the major electrical equipment are the same as for Units No. 1 and No. 2.

Unit No. 4

The following major components were recommended for inclusion in the life extension/maintenance inspection program for the boiler area:

- steam and mud drums;
- superheater and reheater desuperheaters;
- superheater and reheater support and alignment spacers;
- drum, superheater and reheater headers safety valve supports; and
- superheater radiant walls.

Significant items that were noted during a visual inspection of the boilers while in service and a review of generating and maintenance records were:

- No records of superheater and reheater supporting spacers or alignment spacer conditions have been found. The condition of these spacers should be monitored on the next annual outage to determine if any widespread replacement is required due to oxidation or corrosion damage.
- Outer aluminum lagging - in the upper part of the furnace and superheater area between some buckstays the lagging has buckled outwards significantly. The cause of this buckling should be investigated to ensure the pressure parts are not affected.
- The reported high incidence of failure at windbox to burner attachments could require the replacement of the furnace corners if any increase in failure rate occurs (normally, these failures occur at an accelerated rate as a function of the number of unit starts).
- evaluate the extent of data already available versus that required by the EPRI report GS6724, "Condition Assessment Guidelines of Fossil Fuel Power Plant Components", March 1990; and
- provide where possible remaining life/life extension calculations based on this EPRI report.

Assessment by both the outside consultant and the turbine manufacturer indicate that the steam turbine can be operated as required until 2003 under the present maintenance scheme. The IP motor is scheduled to be replaced this year.

In 1988, Hanson Materials Engineering examined locations on the superheater horizontal outlet header and on the main steam lines at the high pressure turbine inlet and outlet for evidence of creep damage. No evidence of creep damage was found. All castings in the steam piping system on this unit have been replaced.

Regular inspections and maintenance of the generator area have been carried out during annual turnarounds. No evidence of major problems or damage have been found. A major inspection of the generator will be carried out during the next annual turnaround.

Findings on the major electrical equipment are the same as for Units No. 1, No. 2 and No. 3.

ADDITIONAL LIFE ASSESSMENT RELATED DATA REQUIRED

The plant reviews noted several outstanding questions and data requirements which bear some relation to Phase I but primarily to the proposed Phase II of this study. These outstanding data include: current lifetime number of startup - shutdown cycles, number and description of excursions from normal operating practices, description of the Unit 1 circulation excursion(s) which necessitated the water wall section replacement, description of the furnace draft excursion which necessitated replacement of the Unit 2 furnace roof, tabulation of any indicated forced coolings of superheater headers and drums beyond the allowable rates indicated by the supplier, any accelerated start-ups of the boilers and steam turbines which would have allowed the pressure part components to exceed the recommended rate of temperature increase, and any air pressure/flue gas draft excursions which affected the boiler pressure parts, the electrostatic precipitators or the duct work.

APPROACH TO PHASE II

While the Phase I work determined that the boilers and steam turbines for all four units can be suitable for base load operation until at least 2003, there are still many gaps in the available specific data.

For Phase II, the scope will include a detailed assessment of the available inspection data and maintenance programs on the Wabamun units to:

- clearly establish the present work being undertaken as "general maintenance" on the major plant equipment;
- evaluate the extent of work actually done versus that recommended in the reports reviewed to date;

FEROPLUG - A TEMPERATURE INDICATOR FOR THE
REMAINING LIFE ASSESSMENT OF HIGH TEMPERATURE
COMPONENTS

by

J.K.L. Lai, C.H. Shek* and B.J. Duggan*
Department of Applied Science
City Polytechnic of Hong Kong
Tat Chee Avenue
Kowloon, Hong Kong.

* Department of Mechanical Engineering,
University of Hong Kong
Pokfulam Road,
Hong Kong

ABSTRACT

The characteristics of ferrite transformation in six casts of specially designed duplex stainless steels have been studied. The materials are useful for making a temperature indicator based on the transformation of the ferrite which can be easily measured by magnetic measurements. This novel temperature indicator can be useful in situations where conventional methods of temperature measurement are not feasible or economical. This device has been given the trade name Ferroplug and is being patented under the auspices of the British Technology Group.

The ferrite content in the duplex stainless steel was found to settle to temperature-dependent steady values after long term exposure to high temperatures. This is a very convenient feature for long term life assessment purposes because the exact exposure time is not required for estimating the temperature.

1. INTRODUCTION

For the energy conversion industry, the deterioration of materials at high temperatures is one major area of concern. Large sums of money are invested by owners and users of high temperature plants to ensure that plant operation remains both economical and safe. The purpose of remaining life assessment is to identify those engineering components which are likely to fail prematurely, and to enable the engineer to make informed decisions on whether to repair, replace or continue operation of the equipment concerned.

Knowledge of the operating temperature

is very important in remaining life assessment. At present, instantaneous temperature indicators such as thermocouples are well known, and although accurate, they require long cables for instrumentation and are sometimes difficult and expensive to use in determining the thermal history of a workpiece. Peak temperature indicators are also known; these consist of a button of material with a specified melting temperature. When this melts it is clear evidence that the workpiece has exceeded the melting temperature of the button. Such an indicator, however is unsuitable in circumstances where unimportant transient temperature fluctuations occur.

Another approach, adopted by Shell Research Ltd., makes use of a temperature history indicator material called Tempplug [1] which depends on the fact that metals and alloys undergo a permanent change in hardness if they are exposed to high temperatures. At a given temperature the amount of change in hardness depends on the time of exposure; conversely, if the time of exposure is known, the hardness change gives an indication of the temperature at which it operated. This approach is powerful in that Tempplugs are versatile, (they can be made in virtually all kinds of shapes from bolts to circlips), durable and cheap. In fact, Tempplugs have been used to measure the temperature in internal combustion engines. The disadvantages of the Tempplug are that hardness tests are best done in the laboratory, as portable hardness tests are less reliable. Laboratory hardness tests are time consuming and semi-destructive.

A different approach invented in Hong Kong and being patented under the auspices of the British Technology Group, uses the transformation characteristics of duplex stainless steels. These steels, so named because they consist of both austenite and ferrite, have the essential property that when they are held at a particular temperature the ferrite transforms to austenite, various carbides and intermetallic phases.

A number of specially designed stainless steels have been produced and the phase transformation characteristics of these alloys have been studied by magnetic measurements and microstructural examination involving optical and electron microscopy. A significant amount of experimental data have been generated on the ferrite transformation behaviour of these duplex stainless steels, some of which have been previously reported [2].

2. EXPERIMENTAL PROCEDURE

2.1 Material and Ageing treatment

Six casts of duplex stainless steel with specially designed compositions (Table 1) were made by the British Steel Corporation. It was received in the form of 5mm thick, 40mm wide strips. Some specimens were solution treated at 1000°C for one hour. In addition, some specimens were cold-rolled 50% and 85% respectively after solution treatment. Specimens were cut to roughly 20mm x 20mm and the thickness depended upon the cold-rolling treatment. Specimens in solution-treated conditions were aged for around 10 to 30 hours in a wide range of temperatures (300°C-700°C in steps of 100°C). The results of this experiment were used as a reference for selecting the temperature range of long term ageing. The long term ageing temperatures were chosen as follows: 400°C, 450°C, 470°C, 510°C and 550°C. Specimens were isothermally aged at these temperatures for about 4000 hours except that the ageing time at 510°C was over 25000 hours.

2.2 Ferrite Measurement

The change in the ferrite content of the specimens was monitored using a magnetic measurement device called the Feritscope [3]. The device was calibrated to give a direct read-out of the volume fraction of ferrite when the ferrite content is between 0-36.5%. For specimens with ferrite content exceeding that limit, the Feritscope can give a reading which increases, though not linearly, with increasing ferrite content up to 80%. The specimens were water-quenched when taken out of furnace and the surfaces were lightly polished to remove the oxide layer before ferrite readings were taken. Ten readings per specimen were taken to calculate the overall average value of each specimen.

3. RESULTS

3.1 Short term ageing results

At ageing temperatures between 300 - 400°C, the ferrite transformation rate is so slow that the ferrite contents of all specimens aged for times up to 30 hours are effectively unchanged. At 500°C, the ferrite content shows a steady decrease with ageing time. At higher temperatures (e.g. 800°C), the transformation appears to reach a stable state in less than 20 hours (see Fig.1).

3.2 Long term ageing results

The ferrite contents of the long-term aged specimens decrease with ageing time and at 510°C the rate of change in ferrite content slows down significantly when the ageing time exceeds about 500 hours (Fig.2). It is clear in Fig.2 that the ferrite contents do not decrease to zero even after 25000 hours of ageing. Instead, they level off to steady values depending on the chemical composition of the material. Other specimens, especially the cold-rolled ones, which have been aged at 550°C or lower temperatures (450°C and 470°C) for over 4000 hours, are also found to exhibit the property of settling to steady values of ferrite content after long term ageing.

The time taken to reach steady values of ferrite content can be shortened significantly by prior cold-rolling the material [4]. Fig.3 shows how the ferrite content varies with time in cold rolled cast C material after ageing at various temperatures between 400°C and 550°C. It can be seen that the ferrite contents of this material settles to temperature dependent values, and it is precisely this property which makes this material a temperature indicator. These graphs can be used as calibration graphs. By comparing the measured ferrite content of a specimen exposed to an unknown temperature with these graphs, it is possible to estimate the temperature.

3.3 Plant trial

Apart from the laboratory experiments, an on-site trial of the Ferroplug temperature

indicator at a power plant in Hong Kong was carried out. Specimens of all six Ferroplug alloys were attached to the surface, underneath the insulation, of the Turbine Main Steam Inlet Loop Pipe of Unit 1 of the Lamma Island Power Station of the Hong Kong Electric Company Limited. The specimens were left there for about 3 years and the accumulated running time of the turbine was about 15000 hours. The temperature estimates made by using the ferrite readings of the six alloys, assuming steady values, were all about 520°C. The information supplied by the Hong Kong Electric Company indicated that the steam temperature inside the loop pipe was 538°C, which differed from the estimated value of 520°C by about 3%. If the effect of temperature gradient along the pipe wall is taken into account, the estimation of temperature with these alloys are remarkably accurate.

4. DISCUSSION

The fact that the ferrite content settles to a temperature-dependent steady value after long term ageing (say, 3000 hours) is a useful property for temperature indication because we do not need to know the accurate accumulated ageing time in order to estimate the operating temperature of the components. For components operating under steady conditions at temperatures between 400-600°C, such as power plant headers and steam pipes, the Ferroplug is particularly suitable for temperature measurement. The value of the ferrite reading alone is sufficient for an estimation of temperature after the steady state has been reached. The cold-rolled materials make better temperature indicators because the time required to reach steady state is shorter [4].

The plant trial result is particularly encouraging and supports the point that the use of this novel temperature indicator in an actual operating environment is quite straight forward and reasonably accurate.

Similar to the Templug, the Ferroplug temperature indicator possesses the advantages of versatility, durability and cheapness. However, it also has the following advantages over the Templug :

- (1) Ferrite measurements can be performed 'in situ' with high reliability using a Feritscope while hardness measurements on Templugs are preferably done in a laboratory.
- (2) Magnetic measurement is much quicker than hardness measurement.
- (3) Magnetic measurement is non-destructive while hardness measurement is semi-destructive. Magnetic measurement can be made at the same location over and over again while hardness measurement cannot be repeated at exactly the same location. Thus a large number of magnetic readings can be taken to improve the reproducibility and accuracy of the measurement.

(4) Duplex stainless steels used in Ferroplugs are more corrosion resistant than the ferritic steels used in Templugs.

Ferroplugs and Templugs can both be made into any convenient shape for the structure being monitored, i.e. it could as easily be made into a bolt screwed into a boiler or a circlip around a pipe. They have no delicate parts and are particularly suitable for use in very hostile environments.

5. CONCLUSION

Based on the significant amount of experimental data on the ferrite transformation in six casts of duplex stainless steels, a novel temperature indicator, Ferroplug, has been developed. It is basically a piece of duplex stainless steel with a specially designed chemical composition, used in conjunction with a Feritscope. Ferroplug not only possesses most of the advantages of a currently available average temperature indicator, Templug, but is superior to it in several aspects.

6. REFERENCES

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Table 1. Chemical composition of the duplex stainless steels

Cast	Elements (wt%)				S	Cr	Mo	Ni	Cu
	C	Si	Mn	P					
A	0.005	0.42	1.57	0.004	0.005	25.1	<0.02	6.00	<0.02
B	0.045	0.44	1.58	0.004	0.004	25.0	<0.02	6.02	<0.02
C	0.007	0.47	1.61	0.004	0.005	25.1	<0.02	7.99	<0.02
D	0.048	0.52	1.56	0.004	0.005	25.6	<0.02	8.06	<0.02
E	0.007	0.49	1.63	0.004	0.005	25.0	<0.02	10.03	<0.02
F	0.048	0.48	1.52	0.004	0.004	25.4	<0.02	10.06	<0.02

FIG.1 AGEING OF SOLUTION-TREATED SPECIMENS AT 800 deg C

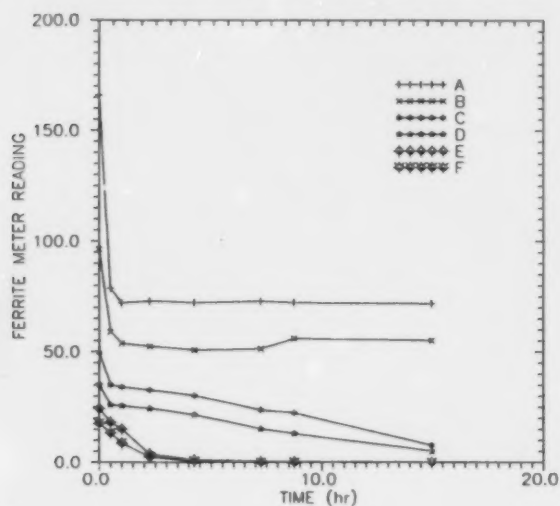


FIG.2 AGEING OF SOLUTION-TREATED SPECIMENS AT 510 deg C

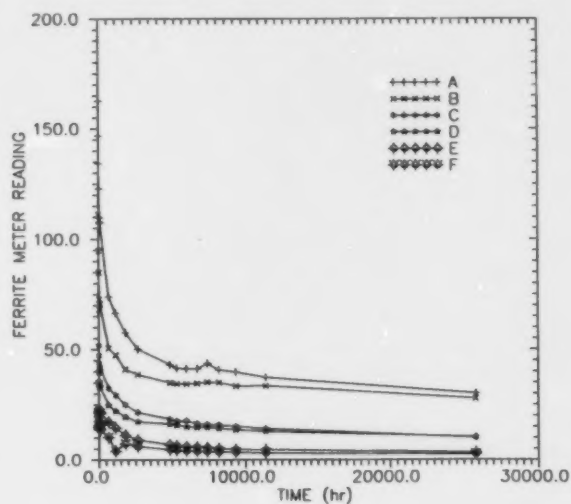
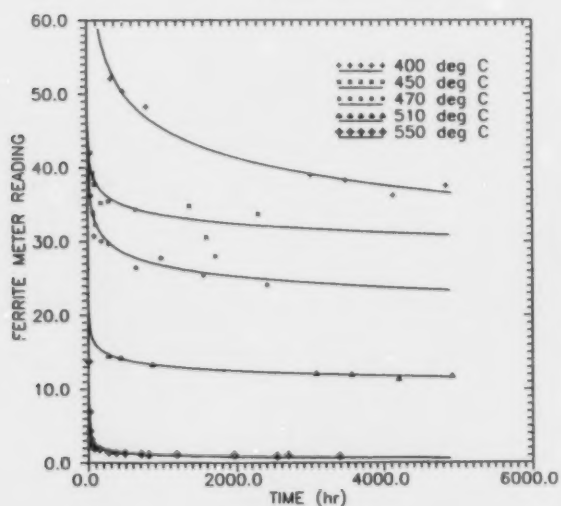


FIG.3 FERRITE CONTENT vs TIME FOR 85% COLD-ROLLED CAST C



DAMAGE AND FUZZY SET RISK ASSESSMENT IN STEAM PLANT

S.K.P. Cheung-Mak
I. Le May

Metallurgical Consulting Services Ltd.
P.O. Box 5006
Saskatoon, Saskatchewan S7K 4E3

ABSTRACT

The various methods of damage assessment in steam plants are reviewed, including microstructural evaluation, nondestructive inspection and creep testing of sample material. An example of estimating the remaining life of steam pipe welds containing cracks, in which crack growth is by creep mechanisms, is discussed.

Modelling techniques are essential to provide a quantitative evaluation of the risk of failure. However, damage assessment is not an exact science. Thus the fuzzy set method of risk assessment is introduced. Most material data for creep are obtained from accelerated creep tests, and the extrapolation of shorter time data to cover longer times at different stresses and temperatures is used in the prediction of remaining life. Inexact knowledge of load and temperature variations in the operation of actual steam plant further complicates overall risk assessment for the entire system.

Estimation of the overall risk of failure in the entire plant using the fuzzy set method of risk assessment requires the identification of critical areas in different components at which damage may have accumulated, together with nondestructive testing, microstructural evaluation and other methods to define the extent of the damage. The importance of the particular component to the operation of the whole system must also be assessed before a judgement can be made to provide a prediction of the remaining life of the plant.

INTRODUCTION

When components fail, the effects of failure (consequences) can be critical [1]. Identification of potential component failures and damage assessment, including their effects on the whole system (in this case a steam plant), should be made periodically after extended periods of operation. It is expensive and often unnecessary to make a risk assessment for all components in the plant to make a sound overall judgement. However, it is important to identify all the critical areas, particularly those that could lead to serious consequences, in which primary failure may lead to several secondary failures in the plant [1].

Critical areas in a high temperature steam plant can be identified largely on the basis of experience and general knowledge of the plant. They include:

- Areas having a design weakness in a component, e.g., local restraint, change in section;
- Areas of physical weakness, e.g., welded joints, obvious cracks or local thinning;
- An area where there is locally high temperature that could lead to concentrated local creep damage;
- Local areas of high heat transfer;
- Regions where deposit build-up on the lower temperature side of a heating surface can lead to excessive metal temperature.

All such areas can become preferentially damaged locally in many instances by creep.

Apart from the obvious significant defects that require immediate repair or replacement, in the more general case, components can be considered to have some remaining safe life, but to have some degree of creep or other damage.

Repair or replacement may involve expensive shutdown of the plant for an extensive period of time. Damage assessment of critical areas, in which there are high local stresses or locally high temperature, should be carried out regularly.

The normal procedure conducted in assessing damage and predicting remaining life in high temperature equipment involves the following:

1. Visual examination;
2. Nondestructive testing such as magnetic particle and dye penetrant testing;
3. Radiographic and/or ultrasonic inspection to evaluate interior crack lengths and depths;
4. Creep damage evaluation or modelling: the evaluation predicts the remaining safe life of the material under the given operating stress and temperature, or the time for a crack to reach a critical length;
5. Microstructural examination;
6. Accelerated creep tests (if required).

The objective of this damage assessment procedure is to determine the extent of the remaining safe life before the component must be replaced. Management and maintenance personnel are interested in quantitative evaluation of risk for critical components and the effect on the whole plant in case of failure; hence, difficulties arise as damage assessment is not an exact science. Damage assessment techniques outlined previously result in an imprecise and inexact life prediction. This paper describes the methodology to provide some quantitative risk assessment for high temperature steam plant where there is a large degree of imprecise knowledge of the parameters such as operating conditions, uncertainties in creep crack growth behaviour and material behaviour.

SYSTEM UNCERTAINTIES IN STEAM PLANT COMPONENTS

Every system contains three types of uncertainties: these are gross, system and random uncertainty. Gross uncertainty includes human errors; system uncertainty relates to the uncertainty or inadequacy of the model used in calculation; while random uncertainty relates to the imprecise knowledge of parameters used in the model.

For elevated temperature steam plant, there is a large degree of system uncertainty, because the modelling process is not exact, and insufficient knowledge exists about the exact mechanisms of damage. In the steam plant where the load and temperature vary with time, together with irregular shutdown periods or intervals, there is random uncertainty in addition to system uncertainty in the predicted life.

The modelling of the residual life of steam pipe joints containing defects can be done using a computer program developed for EPRI [2, 3]. This computer program, PCPIPE, allows prediction of the remaining safe life of elevated temperature steam pipe utilizing creep data and the kinetics of creep crack growth based on the following assumptions:

1. The defect is a through thickness one and computations are 2-dimensional approximations.
2. Creep crack growth behaviour is based on the relation:

$$da/dt = HC_t^q \quad (1)$$

where: da/dt is creep crack growth rate, C_t is a parameter for creep crack tip stress singularity, and H and q are material constants. C_t estimation is based on the assumption of small-scale creep.

3. Plane stress conditions are assumed to compute C_t and J-integral values for crack growth in the axial direction, while material properties used were generated under plane strain conditions.
4. When a component is unloaded, e.g., during shutdown, the residual compressive crack tip stress has not been incorporated in the creep crack growth equation.

The assumptions made in the model generally provide a conservative safe life prediction. The three dimensional model requires time-consuming finite element stress analysis. The cost of development of such analysis is extremely high and the return is comparatively low. The damage mechanism still cannot be fully modelled numerically in an exact sense.

When damage assessment is made based on microstructural examinations of sample material removed from creep damage components, or by nondestructive metallography performed directly on the component, the assessment relies heavily on judgement and personal experience, and on a knowledge of the relation between microstructural changes and the accumulated creep strain versus creep time curve of the specified material

(Fig. 1) [4]. If accelerated creep tests are made, an extrapolation procedure from shorter time creep data at higher stresses and temperatures is required to estimate the probable safe life under the operating conditions [5, 6], or the probable life under the operating conditions experienced is extrapolated, so that a comparison can be made with the predicted life for new material under the same operating conditions. This method has been discussed by da Silveira and Le May [7].

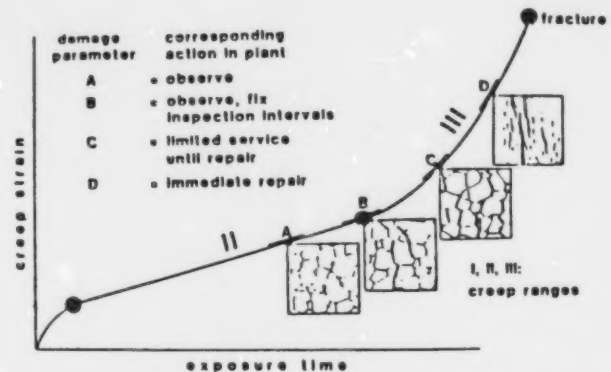


Fig. 1 - The creep damage parameter approach to life assessment, after Neubauer and Wedel [4].

There is a high degree of uncertainty in damage assessment no matter which techniques of life prediction are utilized. To apply reliability theory we require a valid model for the system behavior and the uncertainty needs to be of random nature. Because materials behavior models or constitutive equations for creep are not well defined, the result of probability analysis must be modified, for example, by increasing the variability through multiplication of the output by a random variable. A great deal of judgement is required, and the fuzzy set method is a better alternative method and is introduced here to provide a more complete and better risk assessment of the system.

FUNDAMENTALS OF FUZZY SETS

There is not space in the present paper to provide more than a very brief introduction to fuzzy set theory, and only basic definitions will be provided prior to showing how it can be applied. The interested reader is referred to the sources [8-10] for more detailed background information.

Fuzzy set theory assists a risk analyst in dealing with vagueness caused by the overall complexity of the system and the inherent imprecision. One can consider fuzzy set theory to contain ordinary set theory or that the ordinary set is a special case and is a subset of the fuzzy set. An element is either in or out of an ordinary set, but a fuzzy set has a term named degree of membership level [11], whose value is between zero and one, and which indicates the strength of the belief that an element is in a set.

The fuzzy set is denoted as $A = \{1/1, 2/0.8, 3/0.5, 4/0\}$. This is interpreted as the set A has {1, 2, 3, 4} elements, / is a delimiter, and elements 1, 2, 3 and 4 have degree of membership levels of 1, 0.8, 0.5 and 0 respectively. The set A can be rewritten as $\{1/1, 2/0.8, 3/0.5\}$ since elements with 0 degree of membership are omitted. The element 3 has a degree of membership of 0.5, which indicates that it is not completely in or out of a set.

The fuzzy quantitative computation definitions for fuzzy addition, fuzzy multiplication, and fuzzy division using Zadeh's extension principle [12, 13], will be widely applied on the fuzzy risk analysis. Zadeh's extension principle is a general method for extending functions over the integers to functions over fuzzy subsets based on integers.

Let the universe set be {1, 2, 3, 4, 5, 6, 7, 8, 9} and

$$\begin{aligned} B &= \{1/b(i) \mid 1 \leq i \leq 9\} \\ C &= \{j/c(j) \mid 1 \leq j \leq 9\} \end{aligned}$$

This definition is read as the fuzzy subset B has 'i' elements that have b(i) membership levels, given that i belongs to a set of integers from 1 to 9 inclusive.

The computation definitions are as follows:

$$\begin{aligned} B + C &= \{([i+j]/\min(b(i), c(j))) \mid 1 \leq i, j \leq 9\} \\ B \times C &= \{([ixj]/\min(b(i), c(j))) \mid 1 \leq i, j \leq 9\} \\ B/C &= \{([i/j]/\min(b(i), c(j))) \mid 1 \leq i, j \leq 9\} \end{aligned} \quad (2)$$

In order to compute the membership level of say, 6 in B + C, the minimum membership level of each pair that has a sum of 6 is obtained first and the membership level of the sum 6 is the maximum of all these pairs.

If X is the computed membership level of B + C = 6, then X is as follows:

$$X = \max(\min(b(1), c(5)), \min(b(2), c(4)), \min(b(3), c(3)), \min(b(4), c(2)), \min(b(5), c(1)))$$

These fuzzy computations will be illustrated further in the example presented later.

FUZZY RISK ANALYSIS FOR HIGH TEMPERATURE STEAM PLANT

The fuzzy risk analysis can compute the overall risk for the entire system. The system risk can be decomposed into many components for which their fuzzy risks are estimated. Each component is assessed thoroughly. The estimated probability of failure, the importance of the component to the whole system (the weight) as well as the estimated severity of loss are to be considered in determining the overall system risk.

Figure 2 is a simplified example of a steam plant as a system which has a tree structure. The subtree within the broken lines isolates steam transfer tubes in a steam plant as a system which itself is a subsystem of a steam plant system.

The transfer tube system can be divided into two nodes at its first level: basic design and damage assessment. The starting node in level one is a head node and its descendants are on level two, their descendants in turn being on level three. In the figure, note that the severity of loss and probability of failure are recorded for root nodes at the lowest level. The weight, which evaluates the importance of the failure, is estimated only for the components above the lowest level.

The arithmetic normal weighted mean is extended to be utilized to calculate the fuzzy weighted mean [10]. The fuzzy weighted mean equation is defined as:

$$R = \frac{\sum_{i=1}^n W_i \times R_i}{\sum_{i=1}^n W_i} \quad (3)$$

where: W_i is the severity of loss, R_i is the probability of failure, and n is the number of root nodes.

Let us examine the importance of the weight of the steam tube system. The 'high' weight indicates that it has a significant high risk effect to the whole system. For any system that has an extended history of operation,

the basic design risk estimation is reasonably low and design is assumed to be adequate unless there is drastic design change or unexpected operating conditions. On the other hand, where there are numerous embedded or surface cracks detected, the risk evaluation of creep damage of the components leading to failure would be weighted high. For the case where the plant is newly established, the weights of these two nodes would be opposite to that listed above.

Our fuzzy risk analysis starts from the lowest level. In order to estimate the total risk of a node at level two, the probabilities of failure and the severities of loss for its descendants at level three are all taken into account for fuzzy calculation. Figure 3 shows the parent node at level two and its three root descendants at level three of the steam tube system.

The PCPIPE computer program facilitates the calculation of the predicted safe life of steam tubes containing defects, based on inspection data identifying defects. It is the appropriate damage assessment method for such cases. Table 1 shows the predicted maximum allowable crack depths for three years safe life of different diameters of schedule 160 steam tubes for specified stress at 535°C. The line drawn across in the table indicates the cut off of stress for the projected remaining safe life operation and it has been drawn just above the minimum detectable defect.

As an example, a 356 mm (14 in.) tube in which the predicted longitudinal surface crack depth is greater than 12 mm at 2 kg/mm² operating stress under 533°C would have a safe life of less than three years and would be at high risk. To remain in service, the plant would need either to operate at lower temperature or lower stress, which normally it is not feasible. The alternative would be to replace or repair the tube. Before proceeding with this expensive shut down and replacement, most plants would require further examination than simply to use the conservative modelling procedure. The plant personnel would face the dilemma of which other assessment process should be carried out with an acceptable and effective estimation of safety, considering budget and possible time loss.

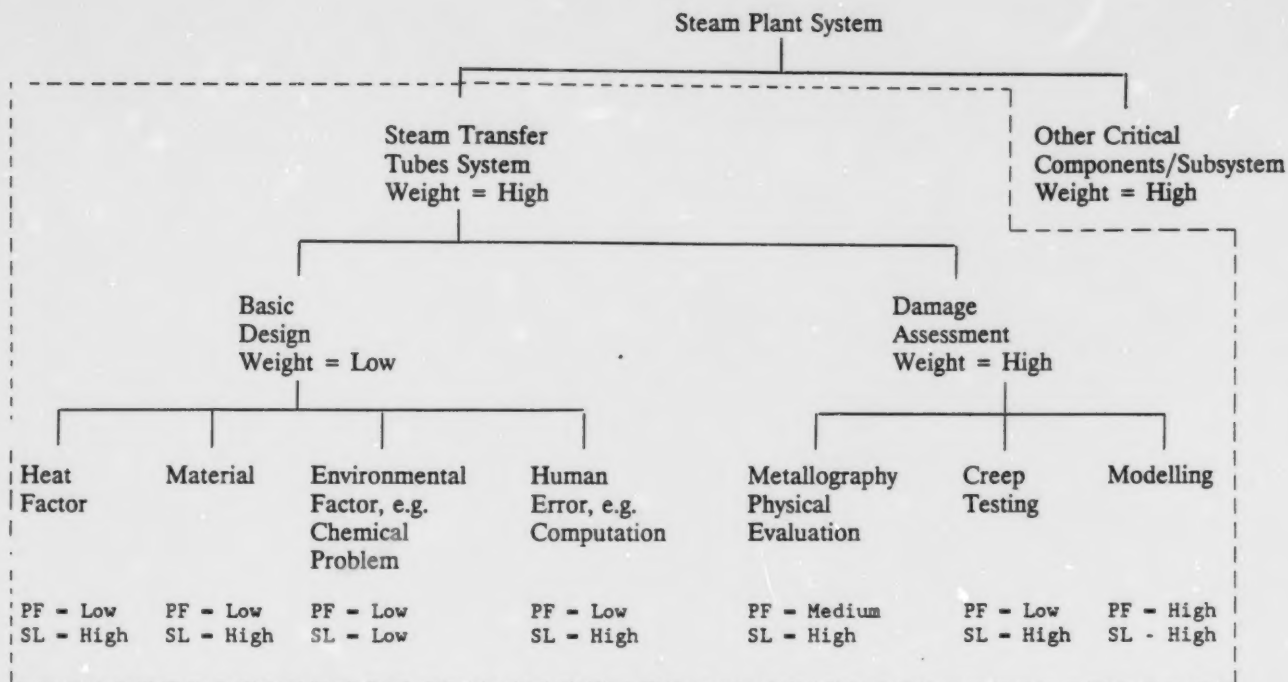
Table 1: Estimated Maximum Crack Depth, a (mm), for Three Years Safe Operation - Longitudinal Surface Flaws

Stress (kg/mm ²)	Tube Nominal Diameter, mm (in.)							
	152 (6)	203 (8)	254 (10)	305 (12)	356 (14)	406 (16)	457 (18)	508 (20)
1	8	11	13	15	17	20	22	25
2	6	8	10	11	12	14	15	17
3	4	5	6	7	8	10	10	12
4	3	3	4	4	5	5	7	7
5	-	-	-	1.25	1.25	1.25	1.25	1.25

In the example shown in Fig. 3, the plant has utilized all three different techniques to evaluate the creep damage. Fuzzy set method summarizes the risk estimate of the level concerned and its tree structure of the system facilitates the identification of the effect of its risk to the whole system.

FUZZY SET RISK ESTIMATION EXAMPLES

The estimation of risk utilizes fuzzy set theory upon which an imprecise situation is modelled. Linguistic variables, whose values are not numbers but words or sentences, such as 'high', 'fairly high', 'medium', 'low to medium' and 'low' allow a more natural



PF = Probability of Failure
SL = Severity of Loss

Fig. 2 - Risk input tree for steam plant system.

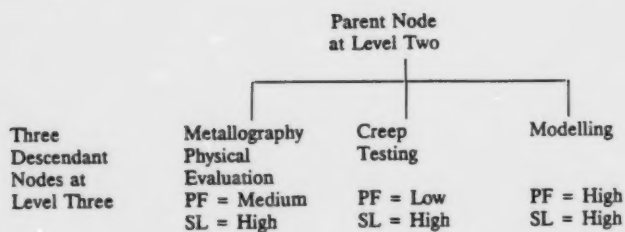


Fig. 3 - Portion of a subtree structure for steam tubes risk analysis example.

specification of values for imprecise concepts [11, 14].

Linguistic terms such as 'low', 'medium' and 'high' can be defined from the set of integers (1, 2, 3, 4, 5, 6, 7, 8, 9). The definitions are arbitrary choices and can be redefined with the meanings to suit the changing system since they are a prior decision made by the system designers.

Let us take Fig. 3 as an example to evaluate the risk of the parent node. In order to simplify the example, fuzzy meanings of 'low', 'medium' and 'high' using (1, 2, 3, 4) as a reduced size of universe set are defined as follows:

Low = (1/1, 2/0.5, 3/0, 4/0)
Medium = (1/0, 2/0.75, 3/0.75, 4/0)
High = (1/0, 2/0, 3/0, 4/1)

Using equations (2) and (3) and the above definitions, the fuzzy risk result can be estimated as illustrated below:

$$\bar{R} = \frac{\sum_{i=1}^3 W_i \times R_i}{\sum_{i=1}^3 W_i} \quad \text{as } n = 3$$

$$\bar{R} = \frac{\text{Medium} \times \text{High} + \text{Low} \times \text{High} + \text{High} \times \text{High}}{\text{High} + \text{High} + \text{High}}$$

Medium x High

$$= \max\left\{\frac{4 \times 1}{\min(1,0)}, \frac{4 \times 2}{\min(1,0.75)}, \frac{4 \times 3}{\min(1,0.75)}, \frac{4 \times 4}{\min(1,0)}\right\} \\ = (8/0.75, 12/0.75)$$

Low x High

$$= \max\left\{\frac{4 \times 1}{\min(1,1)}, \frac{4 \times 2}{\min(1,0.5)}, \frac{4 \times 3}{\min(1,0)}, \frac{4 \times 4}{\min(1,0)}\right\} \\ = (4/1, 8/0.5)$$

High x High

$$= (16/1)$$

High + High + High

$$= (12/1)$$

Medium x High + Low x High

$$= \max\left\{\frac{8+4}{\min(0.75,1)}, \frac{8+8}{\min(0.75,0.5)}, \frac{12+4}{\min(0.75,1)}, \frac{12+8}{\min(0.75,0.5)}\right\} \\ = \max\{12/0.75, 16/0.5, 16/0.75, 20/0.5\} \\ = (12/0.75, 16/0.5, 20/0.5)$$

Medium x High + Low x High + High x High

$$= \max\left\{\frac{12+16}{\min(0.75,1)}, \frac{16+16}{\min(0.75,1)}, \frac{20+16}{\min(0.5,1)}\right\} \\ = (28/0.75, 32/0.75, 36/0.5)$$

Medium x High + Low x High + High x High

$$\text{High + High + High} \\ = ([36+12]\min(0.5,1)) \\ = (3/0.5)$$

The Clements method [15] is used for fuzzy division, of which only quotients of integer type are included, as seen from the above example.

There are different ways to translate the resultant fuzzy set back to the natural linguistic description of the node risk. Here, the Euclidean distance technique [10, 16] is employed and its definition is as follows:

$$\text{distance}(Z, E) = \left[\sum_{i=1}^n (z(i) - e(i))^2 \right]^{1/2}$$

where Z is the computed fuzzy set membership level, E is the assigned fuzzy set with rating membership level and i is the element of the universe.

distance (Z, medium)

$$= \sqrt{(0-0)^2 + (0-0.75)^2 + (0.5-0.75)^2 + (0-0)^2} \\ = 0.79$$

distance (Z, low)

$$= \sqrt{(0-1)^2 + (0-0.5)^2 + (0.5-0)^2 + (0-0)^2} \\ = 1.23$$

distance (Z, high)

$$= \sqrt{(0-0)^2 + (0-0)^2 + (0.5-0)^2 + (0-1)^2} \\ = 1.12$$

From the computed distance, 'medium' is the closest measure of the node risk and one can conclude this node has the medium risk contributed to the whole steam plant.

The whole system risk can then be estimated by climbing through the nodes from lower levels to the head node on level one.

CONCLUSION

Fuzzy set risk estimation proves to be an effective risk analysis that quantitative result can be obtained even when there are various random and system uncertainties in the whole system. This method can extend to any plant damage assessment and it provides a more complete better sound risk analysis. The tree structure of the risk assessment together with fuzzy set theory is a powerful tool to evaluate the combined effects of components in risk assessment for a complex system.

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DAMAGE ASSESSMENT AND LIFE PREDICTION IN HIGH TEMPERATURE PLANT

Iain Le May*
Tito Luiz da Silveira**
Ney Alberto Monteiro Ferreira**

*Metallurgical Consulting Services Ltd., P.O. Box 5006,
Saskatoon, Canada S7K 4E3
and MCS Associates, Inc., Pittsburgh, PA, USA

**Tito Silveira Engenharia e Consultoria Ltda.
Rua Couto Magalhães, Benfica, CEP 20930, Rio de Janeiro, Brazil,
and Faculdade de Engenharia Souza Marques, Rio de Janeiro, Brazil.

ABSTRACT

The nature of microstructural changes and of creep damage that occur in high temperature reformer furnace tubes is described. It is demonstrated that the distribution and morphology of carbides observed in such tubes of HK and HP austenitic steels can be used to determine the distribution of temperature during long time operation. Creep damage can be assigned different levels of damage depending on the microstructural observations. On the basis of such studies of sample tubes the overall condition and remaining safe life of reformer furnaces can be assessed, facilitated by the use of characteristic curves developed from the metallographic features observed.

INTRODUCTION

High temperature plant for the petrochemical industry needs to be inspected and assessed after extended operation in order to determine the degree of damage that has taken place and the remaining safe operating life. One area of particular concern and with which the authors have considerable experience is reformer furnaces. These generally use centrifugally cast tubes of austenitic steel of HK (25 Cr, 20 Ni, 0.4 C) or HP (25 Cr, 35 Ni, 0.4 C) grades. The operating conditions are severe and the maximum wall temperature is generally above 550°C and in some cases greater than 1,000°C.

The life of reformer tubes is limited by creep damage and the normal design process is based on a reference life of 100,000 hours. However, experience has shown that the useful life varies from 30,000 to 150,000 hours. Because of this large scatter, it is important to establish well in advance the optimum time for the planned replacement of old tubes. A proper balance must be made between the longest possible use of the material and the risk of downtime losses caused by repeated failures. This paper outlines a methodology for the estimation of residual life.

THE NATURE OF CREEP DAMAGE AND MICROSTRUCTURAL CHANGES

Creep damage in reformer tubes is detected in the form of apparent aligned pores on grain boundaries, these linking with time to form microcracks and eventually macrocracks [1]. Figure 1 illustrates the nature of the aligned pores which are disclosed in the tube wall approximately one third of the wall thickness from the inner surface after mechanically polishing and etching.

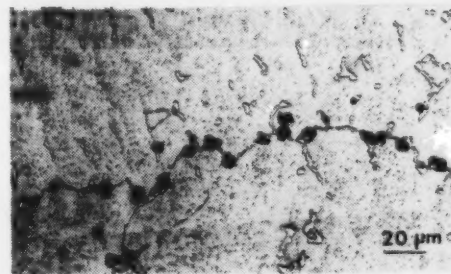


Figure 1 - Creep damage in the form of pores apparent on grain boundaries perpendicular to the principal tensile stress. These are shown up after mechanical polishing followed by etching in Aqua Regia saturated with CuCl_2 . The specimen is a transverse section of a tube of HK 40 from a reformer furnace after 60,000 hours of service.

Figure 2 shows schematically the development of creep damage and the relation between the microstructural observations and the life of the tube [2]. This provides a simple criterion although it involves the sacrifice of a tube or section thereof to allow destructive metallography to be conducted. In particular cases the creep damage may be displaced somewhat further out in the wall thickness if extensive carburization has taken place at the inside.

The observation of isolated voids within the tube wall after polishing and etching is termed Level A damage, and corresponds to approximately one third of the life having been used up (Fig. 2). The number of the

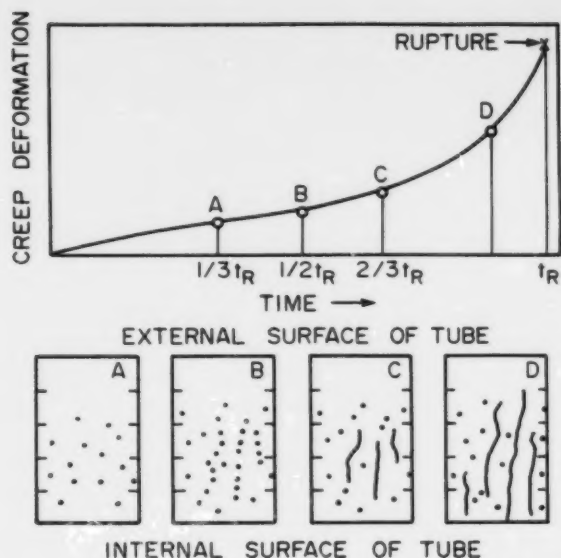


Figure 2 - Schematic showing the classification of creep damage in reformer furnaces. The sketches show how damage can be disclosed after polishing and etching, the corresponding positions along the creep curve being indicated.

lenticular voids disclosed increases as the process of damage accumulation continues: they are generally situated on dendrite boundaries normal to the principal tensile stress present during operation. At about one half of the tube life, Level B of accumulated damage corresponds to the appearance of clearly aligned pores lying in the interior third of the tube wall. Level C of damage, representing two thirds of the life having been consumed, is defined by the presence of microcracks from linking of the aligned voids. These radial microcracks have a longitudinal orientation along the tubes and propagate to the internal surface and may extend halfway through the tube wall. When the microcracks grow in the form of cracks, and where the tube wall free of microcracks becomes less than 40 percent of the total, this characterizes Level D of the accumulated damage. In this case the tube life can be considered close to extinction, requiring its immediate replacement.

Prolonged exposure to high temperature causes alteration of the carbide morphology. Regions of tubes that are exposed to temperatures of up to around 500°C do not display microstructural changes observable with optical microscopy. Very subtle changes involving secondary carbide precipitation adjacent to the primary carbides may be seen as temperature is increased to 600°C. Figure 3(a) illustrates this condition, designated as State I of aging. Between 600° and 700°C, the primary carbides tend to coalesce, losing their platelet morphology and assuming the configuration of compact blocks. Before this process is complete, secondary carbides nucleate in the interior of the austenite matrix. The first secondary carbides precipitate along the edges of the dendrite arms, as may be seen in Fig 3(b), this being characterized as State II. Between 700° and 800°C, the primary carbides transform completely from a eutectic morphology to form compact blocks, and fine secondary carbides are dispersed uniformly throughout the matrix, with individual particles approaching the limit of resolution of the

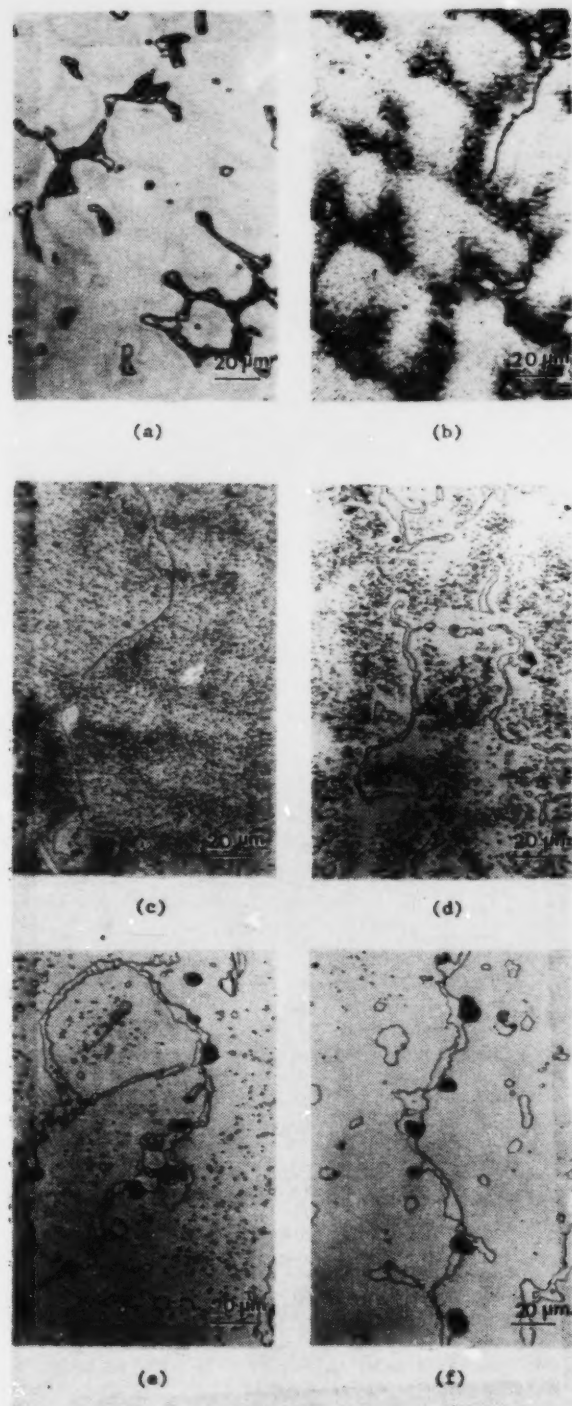


Figure 3 - The stages in aging during extended service at elevated temperature in HK 40 steel. Specimens are transverse ones and were mechanically polished and etched in Aqua Regia saturated with CuCl_2 . They are all taken from close to the inside surface. They were taken from a single tube exposed to 69,000 hours of service. (a) State I; (b) State II; (c) State III; (d) State IV; (e) State V; (f) State VI.

optical microscope. This is shown in Fig. 3(c), the condition being designated as State III. In this state the ductility is greatly reduced, room temperature tensile ductility being between 2 and 5 percent. At temperatures between approximately 800° and 900°C, the morphology of the primary carbides stabilizes and the secondary carbides coalesce. Carbon diffusing to the primary carbides has caused a loss of secondary carbides along the boundaries of the dendrites. This condition, identified as State IV, is shown in Fig. 3(d). If the material is exposed to temperatures between 900° and 1,000°C, larger coalesced secondary carbides are observed, their numbers being significantly less than at lower temperatures. A zone denuded of secondary precipitates may be seen surrounding the primary carbides. These features are illustrated in Fig. 3(e), designated as State V. For operating temperatures above 1,000°C, secondary carbides have disappeared and the austenite matrix has an appearance similar to that of the as-cast condition, Fig. 3(f). This is termed State VI. In this situation the room temperature tensile ductility is between 10 and 15 percent, unless there is significant accumulated damage from creep or other mechanisms.

In Figs. 3 (d to f) the dark pores are representative of creep damage identified after polishing and etching, but these features are irrelevant to the characterization of aging as a result of high temperature exposure, the microstructural changes should not be confused with measurements of damage in the sense that damage is considered to be a cumulative transformation that can affect the functionality of the tubes, or lead to unsafe operation. The transformations described do not affect the functionality of the tubes as has been shown in practice.

In the case of the welds used to join sections of these tubes together, the creep damage develops in the form of apparent pores and cracks lying within the weld metal, in the heat affected zone, or both. These cracks form closer to the inner surface of the tube. Figure 4 illustrates a typical case of damage. The criterion used to judge the extent of damage is the same as for the tubes themselves, involving Levels A to D as illustrated in Fig. 2.

EVALUATION OF ACTUAL FURNACES

As already indicated, the evaluation of the extent of damage in the tubes of reformer furnaces requires the sampling of at least one representative tube. The information obtained can be applied to estimate the probable remaining life of the remaining furnace tubes, and the microstructural features can provide valuable information concerning the operating conditions within the furnace.

Because of the quantity of data obtained in terms of micrographs at different magnifications, an overall picture of the state of a particular furnace may be difficult to appreciate, in terms of creep damage as well as on the basis of operating parameters. The information gathered can be summarized in terms of characteristic curves for the tube. These provide information relating to the operating temperature and the level of creep damage along the representative tube or tubes examined. Figures 5 to 7 show examples from different reformer furnaces.

In Fig. 5 the characteristic curves are shown for a reformer furnace containing 364 tubes of HK 40 steel after 45,800 hours of operation. The welded joints showed apparent voids and microcracks, indicative of damage between Levels B and C. On this basis the remaining safe life was estimated conservatively at around 34,000 hours.

Figure 6 shows the characteristic curves for another



Figure 4 - Creep damage in a weld joining sections of reformer tube. Transverse section adjacent to inside surface. Mechanically polished and etched in Aqua Regia saturated with CuCl_2 .

reformer furnace containing 32 tubes after operation for 65,000 hours. Creep damage disclosed after metallographic preparation had the form of partially aligned lenticular voids between Levels A and B. In this case the remaining life may be estimated to be some 95,000 hours.

The characteristic curves for a reformer furnace with 20 tubes of HK 40 shut down after 69,000 hours of operation are shown in Fig. 7. The furnace was apparently running at a maximum operating temperature of around 1,100°C and the two upper welded joints showed extensive cracking such that they correspond to damage Level D, indicating an end to their useful life. The tube material and lowest weld showed isolated lenticular pores after preparation, a level of damage corresponding to Level B, and an estimated remaining life of 69,000 hours. The tubes were removed from service, the welded joints were reconstructed and the rebuilt tubes were stockpiled for reuse in the future.

It has been found from the metallographic features observed in a certain furnace that the temperature distribution may be much different from those shown in Figs. 5 to 7. For example, a major change in apparent temperature of the order of 300°C was detected over a distance of approximately 2 m in the centre portion of the furnace, probably indicating poor operating conditions in this case.

The distribution of creep damage observed may not correspond directly to the temperature distribution. This is because the stress distribution is complicated: in addition to the stresses caused by internal pressure, there is the stress caused by the weight of the suspended tubes which varies along their length. There may also be some effects from partial support at the bottom of the tubes, as well as thermally induced stresses.

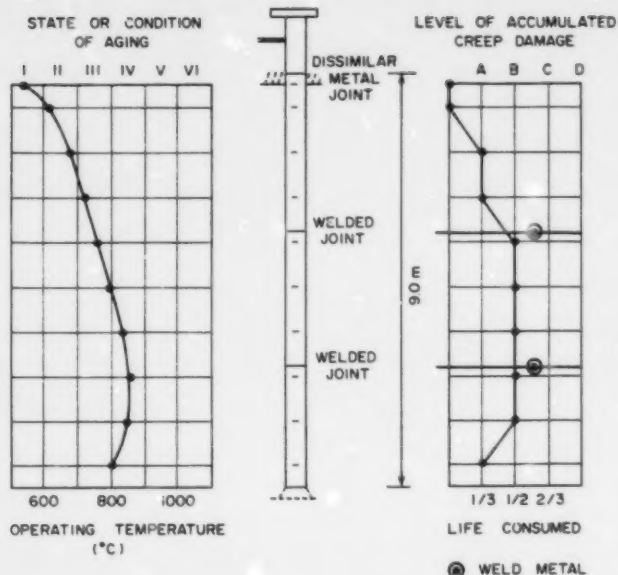


Figure 5 - Characteristic curves developed for a reformer tube of HK 40 removed after 45,800 hours of service. The observed state of aging is shown at the left, with the temperature corresponding to it. At the right, the extent of creep damage at different points along the tube is plotted.

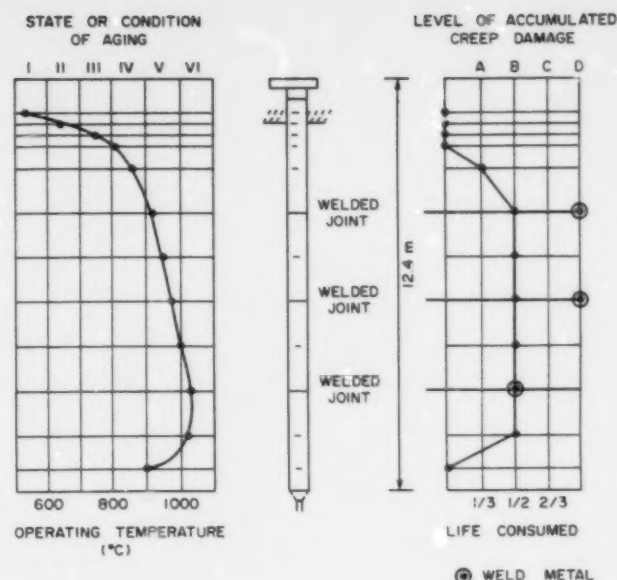


Figure 7 - Characteristic curves developed for a reformer tube of HK 40 removed after 69,000 hours of service. Note that the two upper welds have apparently reached the end of their useful life.

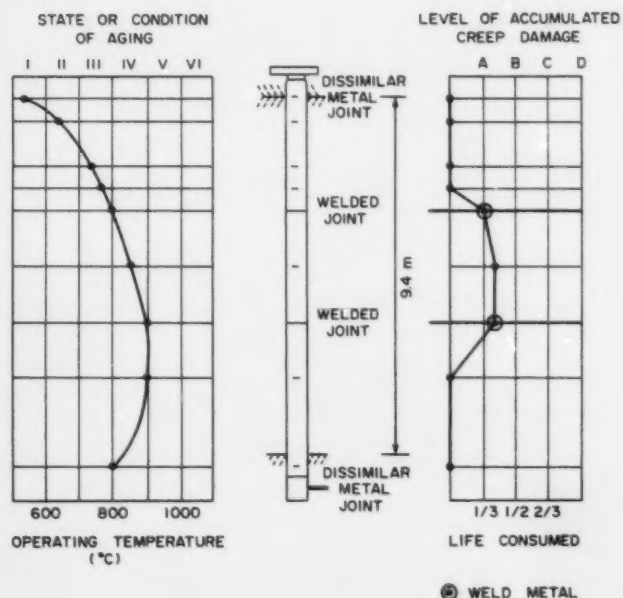


Figure 6 - Characteristic curves developed for a reformer tube of HK 40 removed after 65,000 hours of service.

CONCLUSIONS

Metallographic methods can be utilized to provide information concerning the temperature distribution and the extent of creep damage in high temperature petrochemical equipment. This can provide information concerning the operating conditions and the remaining life of the components.

Two criteria relating to the assessment of temperature and of creep damage in reformer furnace tubes have been presented. In addition the use of characteristic curves for presentation of data obtained has been illustrated.

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BIOGRAPHIES

This section contains the biographies of the authors, invited speakers, and members of the Organizing and Standing Committees of the 11th EAM Symposium.

ALFA, Attahiru S.

Position:

Associate Professor, Department of Mechanical and Industrial Engineering, University of Manitoba, Winnipeg, Manitoba, Canada. R3T 2N2

Qualifications:

B.Sc., M.Sc., Ph.D., P.Eng.

Background

Dr. Alfa obtained his Ph.D. in transportation (with emphasis in Operations Research) from the University of New South Wales, Australia. His research interests are in combinatorial problems and networks.

Archibald, James F.

Position:

Associate Professor,
Department of Mining Engineering
Queen's University, Kingston, Ontario.

Country and Year of Birth:

Canada, 1949

Qualifications:

B.Sc. (Distinction) Mining Eng. 1971, Queen's University
M.Sc. Mining Eng. 1973, Queen's University
Ph.D. Mining Eng. 1976, Queen's University
P.Eng. (Ontario), Registered Consultant, A.P.E.O.
Member CIM, Member CIM Backfill Sub-Committee (Rock Mechanics Group), Member, AIME
Member, Federal Environmental Assessment Review Office (FEARO) Strategic Review Group for Nuclear Waste Management.

Experience:

Dr. Archibald is an Associate Professor in the Department of Mining Engineering at Queen's University. His work experience is primarily in the academic field with some associated private consultation. Dr Archibald's research interest include measurement of radiation levels in underground mines, mine ventilation systems, and structural mine design evaluation.

ARGUE, Shawna L.

Position:

Industrial Systems Engineer
Environmental Division
Saskmont Engineering Ltd.
335 Hoffer Drive
Regina, Saskatchewan
S4N 6E2

Country and Year of Birth:

Canada, 1964

Qualifications:

B.A.Sc. Industrial Systems Engineering 1987,
University of Regina (Co-operative Work
Study Program designation)
Professional Engineer, Association of
Professional Engineers of Saskatchewan
Member, IIE
Member, IEEE (South Saskatchewan Section
Treasurer, 1991-1993)
Member, Regina Business and Professional
Women's Club (Executive member, 1990 -
present)

Experience:

Ms. Argue joined Saskmont Engineering Ltd. in 1988 and has been involved in a variety of industrial projects including electrical design and managing small projects requiring engineering assistance. She joined Saskmont's Environmental Division in 1990 where she has been involved in groundwater monitoring, environmental impact assessments, site assessments and environmental audits. She is currently Saskmont's Environmental Audit Team Leader and has audited industrial facilities in Saskatchewan and Alberta.

Prior to joining Saskmont, Ms. Argue held a contract position with Canada Post Corporation in Regina, Saskatchewan and was with Leon-Ram Enterprises in Yorkton, Saskatchewan.

Bawden, William F.

Position:

Head and Chair in Mine Design,
Department of Mining Engineering
Queen's University, Kingston, Ontario.

Qualifications:

B.Sc. Engineering Geol. 1970, Queen's University
M.Sc. Geotechnical Eng. 1972, U. of Illinois
Ph.D. Geotechnical Eng. 1980, U. of Toronto
P.Eng. (Ontario)

Experience:

From 1972 to 1975, W. Bawden was a project engineer with Acres Consulting Services. From 1975 to 1980 he was a Ph.D. candidate at the U. of Toronto. From 1980 to 1983, he was manager of engineering services for Geotech Ltd of Calgary. In 1983, he joined Noranda Technology where he initiated their geomechanics research program. In 1988, Dr. Bawden was appointed to the chair in mine design at Queen's University. In 1990, he was appointed Head of the Mining Engineering Department.

BAZERGUI, Andre

Position:

Director and Dean, Ecole Polytechnique,
Montreal, Canada, H3C 3A7.

Qualifications:

B.Sc.A. Mech. Eng. 1963, Ecole
Polytechnique, Canada
Ph.D. 1966, Sheffield University, U.K.
Fellow, Applied Mechanics Engineering
Institute of Canada, 1981
Fellow, Canadian Society of Mechanical
Engineers, 1990.

Experience:

Andre Bazergui is a specialist in applied
mechanics (Ph.D. from University of
Sheffield in the United Kingdom) and has
devoted his career to education and research
in this field. He began at the Ecole
Polytechnique in Montreal in 1966 as
assistant professor and became full
professor in 1978 while at the same time
working as Head of the Applied Mechanics
Section until 1985. He is well known for
his expertise in stress analysis and
pressure-tight equipment, and he is also the
author or the co-author of more than 100
scientific and technical papers. Dr.
Bazergui is the Director of the Ecole
Polytechnique since January 1990.

BHOLE, Sanjiwan D.

Position:

Professor and Coordinator, Industrial Systems Engineering,
University of Regina, Regina, Saskatchewan, Canada S4S 0A2.

Country and Year of Birth:

Kenya, 1945.

Qualification:

B. E. Metallurgy. 1969, U. of Poona, India.
M.Sc. Industrial Metallurgy. 1970, U. of Birmingham,
England.
Ph.D. Industrial Metallurgy. 1974, U. of Birmingham, England.
M.B.A. 1978, Cranfield School of Management, England.
P.Eng. (Saskatchewan)
C.Eng. (England)

Experience:

In 1973, Dr. Bhole was a lecturer at the University of
Birmingham. From 1973 to 1977, he was a Research
Investigator at Swinden Laboratories of the British Steel
Corporation, England. After a short assignment (1977-78) at
J.S. University in Iran as an Academic Advisor, Dr. Bhole
joined the Marine Technology Centre at Cranfield Institute of
Technology, England as Research Officer and Centre
Administrator (1979-82). Since 1982, Dr. Bhole has been at
the University of Regina where he is currently Professor and
Coordinator of Industrial Systems Engineering. Dr. Bhole's
research interests include materials selection for process and
manufacturing applications, modeling of structure-property
relationships in materials, and welding. His consulting
activities include developing courses for companies such as
IPSCO, technical assistance to local manufacturers and
contracts with organisations such as SaskPower and
SaskAgriculture.

BOUZID, A. HAKIM.

Position:

Ph.D candidate, Department of Mechanical Engineering,
Ecole Polytechnique of Montreal, P.O. Box 6979,
Station A, Montreal, Quebec, Canada. H3C 3A7.

Country and Year of Birth:

Algeria, 1957.

Qualifications:

B.Sc. Mech. Eng. 1980, U. of Nottingham, U.K.

M.Sc. Mech. Eng. 1981, U. of Leeds, U.K.

Ing. (Quebec).

Experience:

From 1982 to 1984, Mr. Bouzid was employed as a research and development engineer at Sonatrach Oil Company in Algeria. In 1985, he joined an applied mechanic research team in the Energy Research Centre of Algeria, specializing in pressure vessels. In 1986, Mr Bouzid spent a year of sabbatical study in Argentina.

BRIKER, Yevgenia

Position:

Research Officer, Alberta Research Council,
P.O. Bag 1310, Devon, Alberta, Canada.

Country and Year of Birth:

U.S.S.R., 1946

Qualifications:

M.Sc. Chemistry and Chemical Eng., 1969,
University of Chemical Technology of
Leningrad, U.S.S.R.

Experience:

From 1969 to 1970, Y. Briker worked as a Production Engineer at a Plastic Manufacturing plant and from 1971 to 1979 as a Research Scientist at the Research Institute in Lvov, U.S.S.R.

From 1981 to 1985 she worked as a Chemical Technologist at the University of Alberta, Edmonton, in the Chemistry Department (working on bitumen upgrading).

From 1985 until present, she has been working at the Alberta Research Council in the Coal and Hydrocarbon Processing Department, in the area of coal upgrading.

Butt, Stephen

Position:

Undergraduate student. Memorial University of Newfoundland, St. John's, Newfoundland, CANADA

Country and year of Birth:

CANADA

Experience:

Mr. Butt worked for AECL as co-op student, from January 1990 to April 1990.

CHAABAN, Ahmad

Position:

Professor, Department of Mechanical Engineering, Ecole Polytechnique of Montreal, P.O. Box 6079, Station A, Montreal, Quebec, Canada. H3C 3A7

Country and Year of Birth:

Lebanon, 1957.

Qualifications:

B.Sc.A Mech. Eng. 1980, Ecole Polytechnique.
M.Sc.A Mech. Eng. 1982, Ecole Polytechnique.
Ph.D. Mech.Eng. 1985, University of Waterloo
P.Eng. (Quebec).

Experience:

From 1985 to the present time, Dr. Chaaban has been employed at the Ecole Polytechnique of Montreal. He was first appointed Assistant Professor and then Associate Professor in the Mechanical Engineering Department, specializing in pressure vessels. He has supervised 10 M.Sc.A and Ph.D. students, and is the author and co-author of over 20 scientific and technical papers.

CHAN, TIN

Position:

Senior Staff Scientist, Applied Geoscience Branch, Whiteshell Laboratories, AECL Research, Pinawa, Manitoba, Canada, ROE 1L0

Country and Year of Birth:

Hong Kong, 1939

Qualifications:

B.Sc., Mathematics and Physics, 1963, U. of Hong Kong

B.Sc., Special Hon., Physics, 1964, U. of Hong Kong

M.Sc., Physics, 1966, U. of Hong Kong

Ph.D., Physics, 1971, Simon Fraser U.

Experience:

Dr. Chan was employed at the Institute of Earth and Planetary Physics, University of Alberta from 1971 to 1974, at the Cooperative Institute For Environmental Sciences, University of Colorado/NOAA from 1974 to 1975, at AECL Research from 1975 to 1977, and at the Lawrence Berkeley National Laboratory, University of California from 1977 to 1981. Near the end of 1982, after spending a year as Senior Staff Consultant at Intera Environmental Consultants, Inc. in Calgary, he returned to AECL Research where he has been Head of Computation and Analysis, Applied Geoscience Branch since 1985 and Senior Staff Scientist since 1990. He has published over 50 papers related to mathematical and numerical modelling of subsurface processes associated with nuclear waste disposal and has served on many international committees. He is also an Adjunct Professor of Applied Mathematics at the University of Manitoba.

CHEN, Mingyuan

Position:

Assistant Professor, Faculty of Engineering, University of Regina, Regina, Saskatchewan, Canada. S4S 0A2

Qualifications:

B.Eng., M.Eng., Ph.D.

Background

Dr. Chen received Ph.D. in Industrial Engineering from University of Manitoba in 1991. His research interests are in flexible manufacturing systems and networks.

CHEUNG, Steven C.H.

Position:

Associate Professor, Civil Engineering Department, Concordia University, Montréal, Québec, Canada. H3G 1M8

Country and Year of Birth:

Hong Kong, 1943

Qualifications:

B. Eng. 1969, Concordia University, Montréal

M. Eng. 1972, McGill University, Montréal

Ph.D. 1979, McGill University, Montréal

Engineer (Québec)

Member, ASCE, MASCE

Vice Chairman, Engineering Material Division, CSCE.

Experience:

1991-Present: Associate Professor, Civil Engineering Department, Concordia University. Specializing in modelling of contaminant transport, clay barrier design and waste management. He is currently supervising two Masters and two Ph.D. students. He is author and co-author of over 60 scientific and technical papers.

1980-1991: Senior Engineering of AECL Research Vault and Sealing Program, Whiteshell Laboratories, Canada.

CHEUNG-MAK, Peggy S.K.

Position:

Senior Engineer, Metallurgical Consulting Services Ltd., Box 5006, Saskatoon, Saskatchewan, Canada S7K 4E3.

Qualifications:

B.Sc. Mech. Eng. 1982, University of Saskatchewan

M.Sc. Mech. Eng. 1986, University of Saskatchewan

P.Eng. (Saskatchewan)

Experience:

From 1982 to 1983, Peggy Cheung was employed as a research assistant at the University of Saskatchewan. In 1986, she obtained her M.Sc. at the University of Saskatchewan. In the same year, she joined Metallurgical Consulting Services Ltd. She is the author and co-author of over twelve scientific and technical papers.

CLEGHORN, William L.

Position:

Wallace G. Chalmers Professor, Department of Mechanical Engineering, University of Toronto, Toronto, Ontario, Canada M5S 1A4

Country and Year of Birth

Canada, 1953

Qualifications:

B.A.Sc. Mech. Eng. 1975, University of Toronto
M.A.Sc. Mech. Eng. 1976, University of Toronto
Ph.D. Mech. Eng. 1980, University of Toronto
P.Eng. (Manitoba)
Member, CSME, ASEE, SAE

Experience:

Research and teaching in the areas of applied mechanics, mechanisms and vibrations.

COOKE, W. Bruce H.

Position

Professor, Faculty of Engineering, University of Regina, Regina, Saskatchewan, Canada. S4S 0A2

Country and Year of Birth

Canada, 1935

Qualifications

B.A.Sc. Mech. Eng. 1957 University of Toronto
M.Sc. Mech. Eng. 1961 USN Postgraduate School
Ph.D. Mech. Eng. 1972 Queen's University
P.Eng. Saskatchewan

Experience

Dr. Cooke served as an engineering officer in the Royal Canadian Navy from 1957 to 1971. From 1972 to 1979, he worked for the Saskatchewan Power Corporation in the Research and Development Centre. He was Dean of Engineering at the University of Regina from 1979 to 1989.

DA SILVEIRA, Tito Luiz

Position:

Director, School of Engineering, Souza Marques, Rio de Janeiro, Brazil; Professor, Technical Mechanics Dept., Univ. Federal do Rio de Janeiro, Brazil; and Director, Tito Silveira Engenharia e Consultoria Ltda., Rua Couto Magelhaes 744, 20930 Rio de Janeiro, RJ, Brazil.

Qualifications:

B.Sc. Mech. Eng. Universidade Federal do Rio de Janeiro

M.Sc. Metall. Eng. 1975, COPPE/UFRJ

D.Sc. Metall. Eng. 1980, COPPE/UFRJ

Experience:

Professor da Silveira has been on the Engineering Faculty at Souza Marques Engineering School as a professor for more than fifteen years, and is now the Director. He has taught in the Technical Mechanics and Metallurgical Engineering Departments at UFRJ for many years also, and has a number of graduate students in the postgraduate school there (COPPE/UFRJ). Professor da Silveira also heads a consulting company specializing in work for the power and petrochemical industries. He has published many technical and scientific papers and has presented invited lectures at a large number of centres in Latin America, the US, Canada and Europe.

Dayson, C.

Position:

Head, Tribology and Mechanics
Institute for Mechanical Engineering
National Research Council
Vancouver, B.C., V6S 2L2

Country and Year of Birth:

Wales, 1940

Qualifications:

B. Sc. Mech. Eng. 1961, U. of Wales

Ph. D. Mech. Eng. 1965 U. of Wales

Member CSME, I Mech. Eng.

Experience:

From 1961 to 1962 Dr. Dayson was employed by Bristol Siddely Aero Engineers in Bristol, England. In 1966 he joined the Fuels and Lubricants Laboratory of National Research Council in Ottawa where he undertook fundamental and applied studies in Tribology. In 1971 he was transferred to NRC's new Vancouver laboratory where he concentrated on the solution of industrial tribology problems. In 1976 he became the head of this laboratory which expanded to become the present Tribology and Mechanics Laboratory. He is presently also involved in the development of novel mechanical equipment to meet stringent economic, social and environmental requirements.

DE SOUZA, Euler M.

Position:

Professor, Department of Mining Engineering, Queen's University, Kingston, Ontario, Canada, K7L 3N6

Country:

Canada

Qualifications:

M.Sc. Mining Engineering, 1982, Queen's University

Ph.D. Mining Engineering, 1988, Queen's University

Experience:

Dr. De Souza is an Assistant Professor in the Department of Mining Engineering at Queen's University and is responsible for research and teaching in the areas of Rock Mechanics, Mine Ventilation and Mine Hydraulics. He has worked in gold, iron ore and potash mines in Canada and overseas. As a consultant, he has been involved in rock mechanical instrumentation, geotechnical investigations and ventilation design for several mines in Canada and overseas. Dr. De Souza's consulting and research interests include: the development of rock mechanics instrumentation and automatic data acquisition; the numerical modelling of underground rock excavations and structures; field measurements of three-dimensional stress in underground mines; open pit slope stability investigations; dilution ventilation design; mine ventilation system simulation and design; dry drilling investigations; rock drill handle vibration; computer controlled mine dewatering design systems and backfill transportation.

DERENNE, Michel

Position:

Laboratory Manager, Section of Applied Mechanics, Department of Mechanical Engineering, Ecole Polytechnic, Montreal, Canada, H3C 3A7

Qualifications:

B.Sc.A. Mech. Eng. 1967, INSA, France

M.Sc.A. Mech. Eng. 1969, Ecole Polytechnique, Canada

Experience:

Michel Derenne is a specialist in experimental stress analysis and the development of sensors for industrial process. From 1971 to 1975, he was an assistant professor at the INSA Engineering School (France). Between 1976 to 1984, he was the director of his own company dedicated to fine cabinetmaking. From 1984 to 1990, he worked as a research associate at the Applied Mechanics Section of Ecole Polytechnique in Montreal. Since 1990, Mr. Derenne is manager of the Gasket Testing Laboratory of the Mechanical Component Analysis Group (GACM) of Ecole Polytechnique. He is responsible for coordinating the gasket testing activities of this laboratory and reporting for industrial contracts as well as research projects undertaken in this field.

DOOLEY, Barry

Position:

Manager, Boiler and Cycle Chemistry Research in the Office of Exploratory and Applied Research, Electric Power Research Institute (EPRI), Palo Alto, California 94303

Country and Year of Birth:

England, 1944

Qualifications:

B.Eng. Metallurgy 1967, U. of Liverpool
Ph.D. Metallurgy 1970, U. of Liverpool
P.Eng. (Ontario)
Chartered Engineer (U.K.)
Member, Institute of Materials
Executive Secretary, IAPWS

Experience:

From 1970 to 1972, Dr. Dooley was on a post-doctoral fellowship at Queen's University at Kingston, Ontario. From 1972 to 1975, he was a Research Officer in the Materials Division of the Central Electricity Research Laboratories (CERL) of the Central Electricity Generating Board (CEGB) at Leatherhead, Surrey, England. Dr. Dooley joined EPRI in 1984, and until 1988 was a project manager in the Coal Combustion Systems Division where he managed work on fossil plant availability and life extension. He is the author or co-author of over 100 papers and the editor of two books primarily in the areas of metallurgy, power generation, boiler tube failures, cycle chemistry and life extension and assessment of fossil plants.

DUGGAN, Brian J.

Position:

Reader, Department of Mechanical Engineering, University of Hong Kong, Pokfulam Road, Hong Kong.

Country and Year of Birth:

United Kingdom, 1942

Qualifications:

B.Sc. Electrical and Electronics Engineering, 1967, University of Aston, UK.
Ph.D. Industrial Metallurgy, University of Birmingham, UK.
MIM
CEng

Experience:

From 1972 to 1977, Dr. Brian Duggan was a Research Fellow in the Department of Physical Metallurgy of the University of Birmingham, UK. In 1977, he joined the Department of Mechanical Engineering of the University of Hong Kong as a lecturer and has worked on microstructural aspects of plastic deformation and recrystallisation in metals and alloys. He has published over 40 papers in refereed international journals and conference proceedings.

ELLYIN. Fernand

Position:

Director, Advanced Engineered Materials Centre, and Professor, Department of Mechanical Engineering, University of Alberta, Edmonton, Alberta, T6G 2G8

Qualifications:

Dipl. Eng., M.Sc. 1962 University of Tehran
Ph.D. 1966 University of Waterloo

Experience

Dr. Ellyin has over 25 years experience in the development of structural design technology and materials performance for the energy/power industry and the associated safety and reliability of these installations. He has participated as a senior engineer and consultant in design and product development for the oil and gas and pulp and paper industries and for nuclear power plants. His past research and development work includes diverse fields of engineering such as design of pressure vessels and piping systems, materials behavior, fatigue and fracture, creep and high temperature studies of metals, stress corrosion cracking, safety and reliability, computer aided design, and static dynamic response of structures and mechanical systems including interaction effects.

Dr. Ellyin has published over 150 technical papers. and has participated in the writing of ten national standards for nuclear power plants. He has also authored a text book (in French) on the strength of materials and structural analysis. Dr. Ellyin has been an active member of several professional organizations in Canada and abroad.

ELMARAGHY. WAGUIH H.

Position:

Professor, Department of Mechanical Engineering, University of Western Ontario, London, Ontario

Qualifications:

B.Sc., Mech. Eng., 1967, Cairo University
M.Eng., Mech.Eng., 1972, McMaster University
Ph.D., Mech.Eng., 1975, McMaster University

Experience:

After receiving his Ph.D., Dr. ElMaraghy worked at Dofasco as an engineering analyst until 1977, project engineer until 1979, senior development engineer until 1985, and chief engineer until 1986. Dr. ElMaraghy was also an Adjunct Associate Professor at the University of Toronto from 1985 to 1989. He has been at the University of Western Ontario since 1986 where he is presently Professor of Mechanical Engineering and Director of the Design Automation and Manufacturing (DA&M) Research Laboratory. Dr. ElMaraghy has been a member of the Standing Committee of the EAM Symposia since their inception.

FARR, James R.

Position:

Manager, Codes and Regulations, Babcock & Wilcox, Barberton, Ohio 44203

Country and Year of Birth:

U.S.A., 1928

Qualifications:

B.Sc. Civil Eng. 1951, Purdue

P.E. (Ohio)

Fellow, ASME

Experience:

James R. Farr has been involved in the design and construction of fossil and nuclear power generating equipment and pressure vessels for both stationary and marine use since 1952. He has been active with the ASME Boiler and Pressure Vessel Committee since 1958 and is currently chairman of the Subcommittee on Pressure Vessels, Member of the Subcommittee on Design, and a Member of the Main Committee and the Executive Committee. For 30 years, he has been a member of the Pressure Vessel Research Council and the ASME Pressure Vessels and Piping Division and has chaired both.

FERNANDO, C. P. DOMINIC

Position:

Graduate Student,
Industrial Systems Engineering,
University of Regina,
Regina, Saskatchewan,
Canada S4S 0A2.

Country and Year of Birth:

India, 1957.

Qualification:

B. Eng. (Hons)	Mech. Eng., 1979	U. of Madras, India
M. Tech.	Eng. Mechanics, 1989	Indian Inst. of Tech. Madras, India

Experience

From 1979 to 1990, Mr. Fernando has worked with Ms. Bharat Heavy Electricals Limited, India, in various capacities in the areas of erection, commissioning, and design of Power Piping Systems. Presently, he is a student in the Master of Science Programme in Industrial Systems Engineering at the University of Regina, Canada.

FERREIRA, Ney A.M.

Position:

Professor, School of Engineering, Souza Marques, Rio de Janeiro, Brazil; and Engineer, Tito Silveira Engenharia e Consultoria Ltda., Rua Couto Magelhaes 744, 20930 Rio de Janeiro, RJ, Brazil.

Qualifications:

B.Sc. Mech. Eng. Souza Marques Engineering School, Rio de Janeiro.

M.Sc. Metall. Eng. (completing 1992) COPPE/UFRJ

Experience:

A professor at Souza Marques since graduation, Ney Ferreira also undertakes evaluation work on high temperature plant in his capacity as an engineer at Tito Silveira Engenharia e Consultoria Ltda.

Forster, Murray J.

Position:

Graduate Student, Mining Engineering, Rock Mechanics Queen's University, Kingston, Ontario. K7L 3N6

Country and Year of Birth:

Canada, 1957

Qualifications:

B.Sc. Chemistry 1979, U. of British Columbia

B.A.Sc. Geol. Eng. 1982, U. of British Columbia

P.Eng. (Northwest Territories)

EXPERIENCE:

During 1982, M. Forster was an engineer in training for B.C. Hydro, and during 1983, he was a geologist for Falconbridge Ltd. From 1984 to 1988, he was a mine geologist and mine engineer for Giant Yellowknife Mines. From 1988 to 1989, he was a project engineer for Noranda Inc. He is now an M.Sc. student in rock mechanics with a thesis topic on backfill as rockmass support.

GARDINER, William H.

Position:

Manager, Plant Life Extension,
SaskPower, 2025 Victoria Avenue,
Regina, Saskatchewan S4P 0S1

Country and Year of Birth:

United Kingdom, 1936

Qualifications:

LRSC, 1978, London, U.K.

Experience:

From 1953 to 1970, Mr. Gardiner was employed by CEGB in the U.K. in various chemical positions in thermal and nuclear power plants. He has been employed from 1970 to date by Saskatchewan Power Corporation, initially in chemical positions at various locations, and since 1987, as the Manager of Plant Life Extension.

GARLICKI, Andrzej Marian

Position:

Guest Research Scientist, Ground
Transportation Technology Program, Institute
for Mechanical Engineering, National Research
Council of Canada, Ottawa, Ontario

Country and Year of Birth:

Poland, 1923

Qualifications:

1st year, Gen. Eng. 1946, Royal Tech.
University of Turin
D.P.U.C. (Eng.), Mech. Eng. 1951, P.U.C.,
London
Dipl.-Ing., Mech. Eng. 1951, P.B.T.S., London
Ph.D., Mech. Eng. 1976, University of Ottawa
Member, E.I.C.
Member, CSME
P. Eng. (Ontario)

Experience:

For thirteen years following his graduation from the Polish University College in London, Dr. Garlicki was employed with several manufacturing companies in Great Britain and in Canada, gaining experience in heat recovery, steam generation, materials handling, heavy machinery, structures and sheet metal work. In 1964 he relinquished his post as the Chief Engineer of London Concrete Machinery Company in London, Ontario, and joined the staff of then the Ottawa Forest Products Laboratory of the Department of Forestry of Canada, as a Research Officer in the Mechanical Processing Group. In 1979 he joined the National Research Council of Canada and assumed duties of a Research Officer in the then Railway Laboratory of the Division of Mechanical Engineering. In 1988, he was named a Guest Research Scientist at NRC's Institute for Mechanical Engineering. His research activities are reflected in some 55 scientific and technical articles, papers, reports, etc., on various subjects including: power requirements in skidding, debarking, logging with balloons and helicopters, stability and fatigue of bandsaw blades, slicing of wood, and several areas of railway engineering such as train dynamics, brake performance, ride comfort and safety.

Dr. Garlicki is the Vice-Chair of the Standing Committee for Symposia on Engineering Applications of Mechanics (EAM). He was the Chair of the Organizing Committee of the Fifth EAM Symposium, and a member of the Organizing Committees of the 6th, 7th, 8th, 9th, 10th and 11th EAM Symposia. He is the Secretary of the Train Dynamics and Lading Damage Committee.

GARUD, Yogendra S.

Position:

Associate Consultant, S. Levy Incorporated,
3425 S. Bascom Ave., Campbell, CA 95008, U. S. A.

Country and Year of Birth:

India, 1949

Qualifications:

B. Tech. Mech. Eng. 1972, I.I.T., Bombay (India)
M. Tech. Mech. Eng. 1974, I.I.T., Bombay (India)
Ph.D. Mech. Eng. 1981, Stanford University, CA (USA)

Experience:

Dr. Garud joined S. Levy Incorporated (SLI) in 1980 as a consulting engineer and subsequently worked as Project Manager and Associate Consultant. He helped develop and promote the consulting services in the primary areas of structural integrity, fatigue and fracture analysis, stress analysis, and corrosion. His major technical contributions include development and application of quantitative assessment methods for multiaxial fatigue, cyclic plasticity, stress corrosion, and oxidation; he has contributed over 30 published technical articles and reports.

Prior to joining SLI he worked as a research assistant in the Design Division (M.E.) at the Stanford University for 4 years while completing his doctoral work in multiaxial fatigue of metals. In India, he was a Senior Research Assistant in the Design and Production Group (M.E.) in Bombay I.I.T. for about a year; subsequently he worked as a Design and Development Engineer in a private firm where he made significant contributions to the manufacturing and production capabilities of the firm. He left India in 1976 to pursue his continued interest in metal fatigue and applied R&D technology.

GORMAN, DANIEL J.

Position:

Professor, Dep't. of Mech. Eng.
University of Ottawa, Ottawa, Ont.

Country and Year of Birth:

Canada, 1930

Qualifications:

B.Eng. (Mech.) McGill University
M.Eng., Syracuse University
Ph.D., Syracuse University

Experience:

About seven years of experience with Atomic Energy of Canada Ltd., in nuclear reactor operation and research and development work related to nuclear power plants. Member of staff at Univ. of Ottawa, Dep't. of Mech. Eng., since 1970. Major areas of research have been flow induced vibration tube bundles and free vibration analysis of beams and plates.

GRIER, David H.

Position:

Director, Equipment and Machinery Development Division, Saskatchewan Research Council, Saskatoon, Saskatchewan, Canada. S7N 2X8

Country and Year of Birth:

Canada, 1951

Qualifications:

B.E., Agricultural Engineering, 1973, University of Saskatchewan
Various short courses in CAD/CAM, marketing and management.

Experience:

Dave has developed a strong background in the various aspects of product development during his 19 year engineering career. His experience includes machine concept development, user specification development, design, analysis and testing. He specializes in using market research techniques to aid the design process to ensure the needs of the product's user are met. He assists companies in the management of the product development process, requiring marketing, engineering and manufacturing input into the design of new products. He has conducted feasibility studies for national and international clients.

GU, Degiang

Position:

Graduate Research Associate, Department of Agricultural & Bio-resource Engineering, University of Saskatchewan, Saskatoon, Saskatchewan, Canada.
S7N 0W0

Country and Year of Birth:

People's Republic of China, 1954

Qualifications:

B.Sc. Agric. Eng. 1977, Jiangsu Polytechnic University, P.R. of China
M.Sc. Agric. Eng. 1981, Jiangsu Polytechnic University, P.R. of China
Ph.D. Agric. Eng. 1992, Univ. of Saskatchewan, Canada

Experience:

From 1977 to 1979, Mr. Gu was appointed Teaching Associate in the Department of Farm Machinery at the Jiangsu Polytechnic University in China, specializing in Manufacture of Farm Machinery. From 1981 to 1988, He was appointed Lecturer in the Department of Agricultural & Irrigation Engineering at Jiangsu Agricultural University, specializing in Design of Farm Machinery. From 1985 to 1987, Mr. Gu was a Visiting Scholar in the Department of Agricultural Engineering at University of Saskatchewan. He is the author or co-author of 8 scientific and technical papers.

HAGHIGHI, K.

Position:

Associate Professor, Dept. Agricultural
Engineering
PURDUE University, West Lafayette, IN 47907

Experience:

Kamyar Haghighi holds a dual Ph.D. in engineering mechanics and agricultural engineering from Michigan State University. He worked at the Mechanical Engineering Department, Manhattan College, between 1984 and 1986 before joining the Agricultural Engineering Department at Purdue University. He is teaching in the areas of mechanical design and finite element method (FEM). His research interests include integration and networking of mechanical computer-aided engineering tools (FEM, computer graphics, AI, and optimization) to develop expert design systems, development of adaptive and intelligent finite element analysis systems, and nonlinear finite element analysis.

HAMMOUD, Mike

Position:

Service Engineer, Central Region, Babcock
and Wilcox, Cambridge, Ontario

Country and Year of Birth:

Lebanon, 1957

Qualifications:

B.E. Sc. Civil Eng. 1982, U. of Western Ontario
P. Eng. (Ontario)

Experience:

From 1988 to present, Mr. Hammoud was involved in the developing and project management of condition assessment and life extension programs for Canadian Utilities such as Ontario Hydro, Manitoba Hydro and Maritime Electric Company Ltd.

HERAGU, Sundresh S.

Position:

Assistant Professor, Department of Decision Sciences and Engineering Systems, Rensselaer Polytechnic Institute, Troy, New York 12180-3590, U.S.A.

Qualifications:

B.Sc., M.B.A., Ph.D.

Background:

Dr. Heragu previously was Assistant Professor at State University of New York, Plattsburgh. He received Ph.D in Industrial Engineering from University of Manitoba in 1988. His current research interests are in the integration of design and planning activities in manufacturing systems.

HOLTER, Bryan D.

Position:

Engineer, Power Planning, TransAlta Utilities Corporation, 110 - 11th Avenue S.W., Calgary, Alberta, Canada. T2P 2M1

Country and Year of Birth:

Canada, 1958

Qualifications:

B.A.Sc. BioResource Eng. 1981, U. of British Columbia
P.Eng. (Alberta) (A.P.E.G.G.A.)

Experience:

From 1982 to present, Bryan Holter has been employed by TransAlta Utilities Corporation in Calgary, Alberta. From 1982 to 1985 he was employed providing technical and system support for the Energy Management Group at TransAlta. From 1985 to 1989 Bryan worked in TransAlta's Rates & Analysis Section, and was involved in implementing TransAlta's first load research program. In 1989 he moved to his current position in the Power Planning Department where he has developed information on generation supply options, to assist in the strategic planning related to current and future generation resources.

HU, JIH-PERNG

Position:

Project Engineer, Safety Evaluation Group
Reactor Division
Brookhaven National Laboratory
Upton, NY 11973
U.S.A.

Country and Year of Birth:

Taiwan, 1952

Qualifications:

B.S. Aeronautical & Mechanical Engineering,
Cheng-Kung University
M.S. Nuclear Engineering and Engineering
Science, SUNY at Buffalo
Ph.D. Mechanical & Aerospace Engineering, SUNY
at Buffalo
Senior Reactor Operator License (U.S. Nuclear
Regulatory Comm.)
P.Eng. (New York State)
Registered Environmental Professional (U.S.A.)

Experience:

From 1983 to 1988, Dr. Hu was Assistant Professor at Manhattan College in New York City. During this period, he received SRO license from U.S. NRC and was appointed as Chief Reactor Supervisor of the Manhattan College Critical Research Reactor. Between 1988-1990, Dr. Hu obtained P. Eng. and Environmental Professional licenses, and joined Dames & Moore, a worldwide environmental consulting firm, as Environmental Scientist. Since then, he has been employed by Brookhaven National Laboratory, participating in safety analysis and risk assessment for the High Flux Beam Reactor. Dr. Hu has supervised 12 M.S. students in college and has trained 5 reactor operators for the commercial power plant. He is the author or co-author of over 15 scientific and technical papers in the fields of magnetohydrodynamics, aerosol mechanics, waste management, numerical methods, and nuclear technologies.

Hyett, Andrew J.

Position:

Research Associate
Dept. of Mining Engineering
Queen's University, Kingston, Ontario.

Country and Year of Birth:

England, 1963

Qualifications:

B.Sc. Geology 1984, Bristol University
M.Sc. Struct. Geology 1985, Imperial College, U. of London
Ph.D. Rock Mech. 1989, Imperial College, U. of London

Experience:

For the past two years, Dr. Hyett has worked as a research associate at Queen's University, supported on the MRD funded project *Support of Underground Excavations in Hard Rock*.

IGNASIAK, Boleslaw

Position:

Manager, Coal Upgrading and Utilization
Section, Alberta Research Council, P.O. Bag
1310, Devon, Alberta, Canada.

Country and Year of Birth:

Poland, 1938

Qualifications:

M.Sc. Chemistry, 1961, University of Poznan.
Ph.D. Chemistry, 1966, University of Poznan.
D.Sc. Chemistry, 1971, University of Poznan.

Experience:

Dr. B. Ignasiak obtained his M.Sc. (1961) and Ph.D. (1966) in fuel science and technology from A. Mickiewicz University, Poznan, Poland. He came to Canada in 1967 under a National Research Council post-doctoral fellowship program, tenable at the Fuels Research Centre of the Mines Branch, Federal Department of Energy, Mines and Resources. In 1969 he returned to Poland, where for two years he continued his research at the Department of Coal Chemistry and Technology at Mickiewicz University, and there received his D.Sc. degree in fuels science (1971). Soon after, he returned to Canada where he joined the staff of the Alberta Research Council. He has authored and co-authored some hundred papers and patents on fuel science topics.

IRUDAYARAJ, Joseph MK.

Position:

Assistant Professor, Faculty of Engineering
University of Saskatchewan, Saskatoon
Canada S7N 0W0

Country and Year of Birth:

India, 1961

Qualifications:

B.E. Ag. Eng. 1983, India
M.S. Computer Science. 1986, Univ. of Hawaii
M.S. Ag. Eng. 1987, Univ of Hawaii
Ph.D. Ag. Eng. 1990, Purdue University

Experience:

After completing his Ph.D. in 1990 from Purdue University, Dr. Irudayaraj joined the Dept. of Agri. and Bioresource Engineering, Univ. of Saskatchewan as an Assistant Professor. His research interests are in the areas of Mathematical modelling of food systems, Finite Element Analysis and image processing applications in agriculture.

IRVINE-HALLIDAY, Dave

Position:

Associate Professor, Department of Electrical and Computer Engineering, University of Calgary, Calgary, Alberta, Canada. T2N 1N4.

Country and Year of Birth:

Scotland, 1942.

Qualifications:

B.Sc. (Honours) Elect. Eng. 1964, Dundee College of Technology, Scotland

M.Sc Electrical Engineering 1967, University of Aberdeen, Scotland

Ph.D. Electrical Engineering 1969, University of Aberdeen, Scotland.

MIEE (U.K.)

MIEEE (U.S.A.)

SMIRE (AUST.)

C.Eng (U.K.)

P.Eng (Alberta)

Experience:

1964-65 Engineer in Training Submerged Repeater Department at STC, London, U.K.

1970-74 Member of Scientific staff in Digital Switching and Transmission Systems Engineering, BNR, Ottawa.

1974-78 Senior Engineer in Engineering Training, AGT, Edmonton.

1979-80 Technical Engineer Transport Division for Highland Regional Council, Kingussie, Scotland.

1980-82 Lecturer in Electrical Engineering Department at Victoria University of Technology, Melbourne, Australia.

1982- Associate Professor in Electrical and Computer Engineering at the University of Calgary.

Main research interests are in fibre optic communications and sensors, optical switching and high performance LAN protocols.

IWANIW, Marie A.

Position:

Associate Professor, Faculty of Engineering, University of Regina, Regina, Saskatchewan, Canada S4S 0A2

Country and Year of Birth:

Canada, 1951

Qualifications:

B.Sc. Eng. Physics, 1973, Queen's University

M.Sc. Mech. Eng. 1975, Queen's University

Ph.D. Mech. Eng. 1983, Queen's University

P.Eng. (Saskatchewan)

Member, ASAE/CSAE

Experience:

From 1984 to 1988, Dr. Iwaniw was an Assistant Professor in the School of Engineering of the Ontario Agricultural College at the University of Guelph, Ontario. In 1988, she was appointed Associate Professor in the Faculty of Engineering at the University of Regina, Saskatchewan. Her research interests include the application of fluid mechanics principles to the flow in naturally-ventilated livestock structures.

JASKE, Carl E.

Position:

Senior Group Leader, Materials Engineering and Research, Cortest Columbus Technologies, Inc., Columbus, Ohio 43235

Country and Year of Birth:

U.S.A., 1943

Qualifications:

B.Sc. Liberal Arts & Sciences, 1966, U. of Illinois

B.Sc. General Engineering, 1966, U. of Illinois

M.Sc. Theoretical and Applied Mechanics, 1967, U. of Illinois

Ph.D. Metallurgical Engineering, 1984, Ohio State

P.Eng. (Ohio)

Fellow, ASME

Experience:

From 1967 to 1990, Dr. Jaske worked at Batelle in Columbus, Ohio. In November 1990, he joined Cortest Columbus Technologies, Inc. (CC Technologies) as a Senior Group Leader in Materials Engineering and Research. He is leading work in the areas of fatigue, corrosion-fatigue, creep, creep-crack growth, high-temperature materials properties, in-service aging of materials, and plant-life extension and assessment. He has developed and implemented fracture-mechanics approaches for dealing with creep, thermal-fatigue, low-cycle fatigue, and creep-fatigue crack growth in structural integrity assessment and life production, especially for pressure vessels and piping that are in service. Also, he has developed special-purpose computer codes for the remaining life assessment of welded steam pipes and reformer furnace tubes.

JAYASHANKAR, M.

Position:

Graduate Student, Industrial Systems Engineering, University of Regina, Regina, Saskatchewan, Canada S4S 0A2.

Country and Year of Birth:

India, 1955.

Qualification:

B. E. (Honours), Mech. Eng. 1977, U. of Madras, India.

M. E. Production Eng. 1991, Anna University, India.

Experience:

From 1978 to 1991, Mr. Jayashankar was employed at Bharat Heavy Electricals Limited, Madras, India, as Power Piping Design Engineer. Presently, he is a student in the Master of Science Programme in Industrial Systems Engineering at the University of Regina, Canada.

JIN, YEE-CHUNG

Position:

Assistant Professor, Faculty of Engineering,
University of Regina, Regina, Saskatchewan,
CANADA.

Country and year of Birth:

Taiwan, ROC, 1953.

Qualifications:

B.S.E. Hydraulic Eng. 1976, National Cheng-
Kung University
M.Sc. Civil Eng. 1982, Auburn University
Ph.D. Civil Eng. 1990, University of Alberta
P.Eng. (Alberta)

Experience:

Dr. Yee-Chung Jin is an assistant professor of Engineering at University of Regina. Prior to this he worked for Atomic Energy of Canada Ltd. as hydrogeology modeller for two years. His current research is in the area of numerical simulation for groundwater flow and contaminant transport in aquifer system and computer modelling of the flow in open channels. He is the author or co-author of 6 scientific and technical papers.

KAMINSKI, Tony L.

Position:

Senior Design Engineer, Equipment and
Machinery Development Division, Saskatchewan
Research Council, Saskatoon, Saskatchewan,
Canada. S7N 2X8

Country and Year of Birth:

Canada, 1939

Qualifications:

Ph.D., Engineering, 1965, Michigan State
University
M.S., Engineering, 1963, University of
Saskatchewan
B.E., 1961, University of Saskatchewan
Registered Professional Engineer

Experience:

As well as a Ph.D. in Engineering, Tony has over 27 years of good practical "hands-on" design engineering experience gained primarily in industry by working on conceptual design, feasibility and marketing studies, engineering analysis, computer modelling, development and evaluation of machinery. From 1965 to 1985 he was employed by the Advanced Technology Center of Allis Chalmers Mfg. Co. in Milwaukee, Wisconsin. Since 1985 he has been with SRC doing engineering work primarily for manufacturers. Valuable professional contacts have been established with manufacturers and suppliers of machinery and components, research and development agencies and institutions across North America.

KESTNER, J. Scott

Position:

Senior Supervising Engineer, Monenco Inc., 801 - 6th Avenue S.W., Calgary, Alberta, Canada. T2P 3W3

Country and Year of Birth:

Canada, 1943

Qualifications:

B.Sc. Mech. Eng. 1965, U. of Alberta

Experience:

From 1965 to 1974 Mr. Kestner was employed by Montreal Engineering Company Ltd. (now Monenco Inc.) at junior, intermediate and senior engineering levels on the design and construction of thermal generating stations and a heavy water plant. From 1974 to 1977 he worked for Associated Engineering Services as a senior engineer then Mechanical Department Manager in general industrial facilities design. In 1978 to 1980 he was employed by Simmons Drilling Ltd. and worked in the oil well drilling and seismic fields. Mr. Kestner returned to Monenco in 1980 as a senior supervising engineer primarily in the area of thermal generating plants.

KHAIR, Khairy R.

Position:

Research Scientist/Engineer
Applied Geoscience Branch, AECL Research Pinawa,
Manitoba, Canada. ROE 1L0

Country and Year of Birth

Egypt, 1948.

Qualifications

B.Sc. Mech. Eng. 1971, Cairo U., Egypt
Diploma of Higher Studies, Mech. Eng. 1977, Cairo
U., Egypt
M.Sc. Mech. Eng. 1984, U of Colorado, Boulder.
Ph.D. Mech. Eng. 1989, U of Colorado, Boulder.

Experience:

From 1971 to 1978, Dr. Khair worked for ELTRAMCO in Cairo, Egypt as industrial engineer. In 1978, he moved to the project engineering section and became the section head in 1980. He prepared the feasibility studies for new production lines and supervised engineers and technicians to organize the civil engineering activities for the construction of these lines. In 1984, he obtained his M.Sc. in the area of heat transfer. In 1986, he worked for the Cooperative Institute for Research in Environmental Science as research assistant. His research focused on studying the stress accumulation before moderate earthquakes. In 1989, Dr. Khair obtained his Ph.D. in the area of solid dynamics. He joined AECL Research in Pinawa, Manitoba, Canada as research engineer specializing in numerical modelling of rock behaviour and fluid flow in porous medium. Dr. Khair is the author and the co-author of 14 scientific and technical papers.

KIZHATIL, Revi K.

Position:

Graduate Student,
Industrial Systems Engineering,
University of Regina, Regina,
Sask., Canada. S4S 0A2

Country and Year of Birth:

India, 1964

Qualifications:

B.Tech. Mech. Eng. 1986, REC Calicut, India.
M.Tech. Eng. Mechanics 1988, Indian Inst. of Tech., Madras, India.
Ph.D Ind. Sys. Eng. 1988-Present, U. of Regina, Regina, Canada.

Research Interests:

Mr. Kizhatil is presently enrolled in a doctoral program at the University of Regina. His areas of interest include plasticity, creep, fatigue and fracture mechanics. His thesis is in the area of developing simplified methods for inelastic analysis of pressure components. Mr. Kizhatil has to his credit nine technical publications.

KUSHWAHA, R. Lal

Position:

Professor, Department of Agricultural and Bioresource Engineering, University of Saskatchewan, Saskatoon, Saskatchewan, Canada. S7N 0W0

Country and Year of Birth:

India, 1940

Qualifications:

B.Sc. Ag. Eng. 1961, University of Allahabad
M.Sc. Ag. Eng. 1964, University of Toronto
Ph.D. Ag. Eng. 1967, U. of Saskatchewan
P. Eng. (Saskatchewan)
Member: ASAE, CSAE, SAE, ISTVS, STLE, ASM

Experience:

From 1961 to 1963, Dr. Kushwaha was employed as a Research Engineer/Lecturer at the Allahabad Agricultural Institute, India. After his studies in Canada, he worked for the Ford Foundation in New Delhi, India (1968-69) under the Indian Agricultural Development Program. In 1969, he was appointed Associate Professor in the Division of Agricultural Engineering, Indian Agricultural Research Institute, New Delhi. In 1972, he came to Canada and was employed as In-Charge Quality Control in the Automotive Parts Division of Bundy of Canada, in Guelph, Ontario. In 1974, Dr. Kushwaha returned to the University of Saskatchewan to work as a Professional Research Associate in the Department of Agricultural Engineering. He was appointed as a Research Scientist in 1980, Associate Professor in 1986, and Professor in 1991, specializing in agricultural mechanization systems, tillage, traction and terramechanics, crop production and harvesting equipment and tribology in agriculture. He has supervised five M.Sc. and Ph.D. students and is the author or co-author of over 80 scientific and technical papers.

LAI, Joseph K. L.

Position:

Principal Lecturer, Department of Applied Science, City Polytechnic of Hong Kong, Tat Chee Avenue, Kowloon, Hong Kong.

Country and Year of Birth:

Hong Kong, 1952

Qualifications:

B.A. Physics, 1974, University of Oxford, UK.
M.A. 1978, University of Oxford, UK.
Ph.D. Materials Science, City University, UK.
MIM
MIMechE
CEng

Experience:

From 1974 to 1985, Dr. Joseph Lai was employed as a Research Officer at the Central Electricity Research Laboratories at Leatherhead, Surrey, UK. His research area is properties of stainless steels for power plant applications. In 1984, he was appointed Project Leader of the Remaining Life Study Group and member of the Remanent Life Task Force of the Central Electricity Generating Board. Dr. Lai returned to Hong Kong in 1985 to a teaching position in the City Polytechnic of Hong Kong and he is now a Principal Lecturer in the Department of Applied Science. Dr. Lai is currently supervising 4 MPhil/PhD students, and is the author or co-author of over 40 scientific and technical papers.

Lausch, Peter

Position:

Research Assistant & Guest Lecturer,
Department of Mining Engineering
Queen's University, Kingston, Ontario.

Country and Year of Birth:

Canada, 1957

Qualifications:

Cet. Civil Eng. 1980, Ryerson Polytechnical Institute of Tech.
B.Sc. Mining Eng. 1984, Queen's University
M.Sc. Mining Eng. 1989, Queen's University

LE MAY, Iain

Position:

President, Metallurgical Consulting Services Ltd., P.O. Box 5006, Saskatoon, Saskatchewan, Canada, S7K 4E3, and President, MCS Associates, Inc., Pittsburgh, PA, USA.

Qualifications:

B.Sc. Mech. Eng., 1957, University of Glasgow
A.R.C.S.T. Mech. Eng., 1957, University of Strathclyde
Ph.D. Materials, 1963, University of Glasgow
P.Eng.
Fellow, ASM

Experience:

Dr. Le May was on the faculty of the University of Saskatchewan from 1963 to 1985 with responsibility for materials and metallurgy teaching and research. Since 1985 he has been a full time consultant. He founded Metallurgical Consulting Services Ltd. in 1978, and has acted as a consultant for companies in North and South America, Europe and Japan. Dr. Le May has authored over 200 technical papers and several books.

LEE, C.F.

Position:

Manager, Civil Engineering and Architecture Department, Ontario Hydro

Country & Year of Birth:

China, 1945

Qualifications:

B.E.Sc. 1968, University of Hong Kong
M.E.Sc. 1970, University of Hong Kong
Ph.D. 1972, University of Western Ontario

Experience:

1975-Present: Geotechnical/Civil Engineering Practice, Ontario Hydro. Highlights include: Radioactive waste disposal; Nuclear power plant siting; Hydroelectric power plant; site investigation/evaluation; Earthquake engineering; Dam Safety Assessment.

1972-1975: Assistant Professor, Geotechnical Engineering), University of Western Ontario.

LIANG, Ming

Dr. Liang is an Assistant Professor of Mechanical Engineering at the University of Ottawa. His current research focuses on the tooling and planning problems in flexible manufacturing systems. Dr. Liang is a member of SME and IIE.

MACFARLANE, William A.

Position:

Manager of Special Projects, SNC -LAVALIN, Vancouver, British Columbia, Canada.

Country and Year of Birth:

Scotland, 1944.

Qualifications:

B.Sc. Mech. Eng. 1966, University of Glasgow.

P.Eng. Alberta.

P.Eng. British Columbia.

Experience:

From 1966 to 1974 Mr. Macfarlane worked in the Electricity Supply Industry in the U.K. firstly with the boilermaker, Babcock and Wilcox Ltd. and then with the utility, the South of Scotland Electricity Board. In 1974 he relocated to Canada and until the present time has been involved mainly in Consulting in the Thermal Power, Pulp and Paper and Oil Sands Industries.

MARCHAND, Luc

Position:

Assistant Professor, Section of Applied Mechanics, Department of Mechanical Engineering, Ecole Polytechnic, Montreal, Canada, H3C 3A7

Qualifications:

B.Sc.A. Mech. Eng. 1975, Ecole Polytechnique, Canada
M.Sc.A. Mech. Eng. 1982, Ecole Polytechnique, Canada
Ph.D. Mech. Eng. 1991, Ecole Polytechnique, Canada

Experience:

Luc Marchand is a specialist in the field of gasket testing and experimental stress analysis. From 1976 to 1979, he worked at Canadair (Montreal) as a stress engineer involved in the design of aircraft structures. From 1981 to 1987, as a research associate at the Department of Mechanical Engineering of Ecole Polytechnique in Montreal. he was responsible for design and complementation of gasket testing equipment. Since 1991, Dr. Marchand is assistant professor at Applied Mechanics Section of Ecole Polytechnique where he teaches experimental stress analysis techniques. Much of his research work is devoted to the study of long term performances of gaskets at elevated temperature and the behavior of bolted flanged connections at room temperature.

MARRIOTT, Douglas L.

Position:

Associate Professor, Mechanical and Industrial Engineering, University of Illinois at Urbana-Champaign

Qualifications:

B.Sc., 1961, University of Natal
M.Sc., 1963, University of Natal
Ph.D., 1966, Cambridge

Experience:

Douglas L. Marriott was a lecturer at the University of Natal (1966-68) and at Liverpool University (1969-76). He has also been a visiting lecturer at Chalmers University in Gothenburg, Sweden (1972) and University of Rhodesia (1978). Dr. Marriott was Head of the Future Studies Group and Assistant Director of the Licensing Branch, South African Atomic Energy Board (1976-79). Since 1980, Dr. Marriott has been in the Mechanical and Industrial Engineering Department, University of Illinois at Urbana-Champaign where he is currently an Associate Professor. Dr. Marriott has completed a wide array of consulting activities with many organizations including the US Army Corps of Engineers, Oak Ridge National Laboratory, UK Atomic Energy Authority, Sunstrand Aviation and Procter and Gamble.

Mercer, Robert A.

Position:

Graduate Student,
Department of Mining Engineering
Queen's University, Kingston, Ontario.

Country and Year of Birth:

Canada, 1965

Qualifications:

B.Sc. (Honours) Geological Eng. 1988, Queen's University

Experience:

From 1988 to 1990, Mr. Mercer worked as a Rock Mechanics Engineer for Vaal Reefs Exploration and Mining Co. Ltd. in Klerksdorp, South Africa. In 1991, Mr. Mercer returned to Queen's University to work towards a Master's in Mining Engineering.

MIKETINAC, M.J.

Position:

Professor, Dept. of Mathematics and
Statistics, University of Saskatchewan,
Saskatoon, S7N 0W0, Canada.

Country of Birth:

Croatia

Qualifications:

B.Sc. and M.Sc. University of Zagreb, 1964
and 1964

Ph.D. University of North Carolina, Chapel
Hill, 1968

Experience:

He has supervised 6 Graduate Students and
published about 30 papers. His main
interest is Numerical Solution of Partial
Differential Equations Arising from
Technological Applications.

MITCHELL, Robert J.

Position:

Professor, Department of Civil Engineering, Kingston,
Ontario, Canada, K7L 4V2

Qualifications:

B.A.Sc. University of Waterloo
Ph.D. Cambridge University
P.Eng. (Ontario)

Experience:

Dr. Mitchell obtained his Ph.D as an Athlone Fellow and a N.R.C. scholar. He has worked in the mining industry and with government agencies in geotechnical research, water resources and materials engineering. He is currently chairman of the University Liaison committee of the Canadian Geotechnical Society.

MONAGHAN, Paul P.

Position:

Statutory Lecturer
Dept. Mechanical Engineering
University College
Galway, Republic of Ireland

Place and Year of Birth:

Northern Ireland, 1953

Qualifications:

B.Sc. Mech. Eng. 1975, Queen's University,
Belfast, Northern Ireland
M.Sc. Mech. Eng. 1976, Queen's University,
Kingston, Canada
Ph.D. Mech. Eng. 1988, Queen's University,
Kingston, Canada

Experience:

From 1976 to 1977, Mr. Monaghan was a research assistant at Trinity college, Dublin. From 1977 to 1981 he was employed as an Energy Engineer by Chorley and Bisset Ltd. of London, Canada and from 1981 to the present he has been Statutory Lecturer at University College, Galway, Ireland. He is the author of many papers on various aspects of energy use and frost formation.

MORSHED, Ahmed Hisham

Position:

Assistant Lecturer, Electronics and Comp. Engineering Department, Faculty of Engineering, Ain Shams University, Cairo, Egypt and a Graduate Student at The Electrical and Computer Engineering Department, University of Calgary, Calgary, Alberta, Canada. T2N 1N4.

Country and Year of Birth:

Egypt, 1961

Qualifications:

B.Sc. Electrical Engineering 1984, Ain Shams University

M.Sc. Electrical Engineering, Ain Shams University

St.M IEEE

St.M OSA

P.Eng (Egypt)

Experience:

From 1984 to 1989, Mr. Morshed worked as a Teaching Assistant at the Electronics and Comp. Engineering Department, Faculty of Engineering, Ain-Shams University. In 1989, he was appointed Assistant Lecturer at the same department. From 1990 till now he has been a Graduate Student and Teaching Assistant at the Electrical and Computer Engineering Department, University of Calgary. His research at Ain Shams University included investigation of optical wave propagation and coupling in curved and compound etalon semiconductor lasers. He is currently involved in research on optical fiber sensors and sensor networks.

MOURADIAN, Ara

Position:

Research Assistant, Civil Engineering Department, Concordia University, Montréal, Québec, Canada. H3G 1M8

Country and Year of Birth:

Lebanon, 1962

Qualifications:

B. Civil Engineering 1984, American University of Beirut

Associate Member, CSCE.

Experience:

1984-1990: Design Engineering, I&M Design, Architecture and Contracting, Beirut

OOSTHUIZEN Patrick H.

Position:

Professor
Dept. of Mechanical Engineering
Queen's University
Kingston, Ontario
Canada K7L 3N6

Country and Year of Birth:

South Africa, 1938

Qualifications:

B.Sc. Mech. Eng. 1959, University of Cape Town,
South Africa
M.Sc. Mech. Eng. 1963, University of Cape Town,
South Africa
M.A.Sc. Aero. Eng. 1967, University of Toronto,
Canada
Ph.D. Mech. Eng. 1968, University of Cape Town,
South Africa

Experience:

Professor Oosthuizen has been at Queen's University since 1968. He is the author of more than 250 technical papers concerned with various aspects of Heat Transfer, Fluid Mechanics and Energy Supply and Use.

OPSETH, Arthur L.

Position:

Assistant Dean
and Professor Systems Engineering
Faculty of Engineering
University, Regina, Saskatchewan S4S 0A2

Qualification:

B.E. 1959, M.Sc. 1961 Mechanical Engineering
University of Saskatchewan

Experience:

Prof. Opseth has been a member of the Faculty of Engineering since 1974. Before that he had ten years experience with Saskatchewan Government in Computer Programming and Systems Design. In addition he spent three years in industry as a design engineer working in fatigue and the design of equipment for armoured personnel carriers. He grew up on a Saskatchewan farm and has always been involved in the use of farm equipment and also the maintenance and repair of this equipment.

His duties at the University of Regina have involved teaching Graphics, Statics and Dynamics, Strength of Materials, Introduction to Engineering Design and Engineering Law and Professionalism. He served for four years as Cooperative Education Coordinator for the Faculty, one year as Acting Dean and now five years as Assistant Dean. He has been active with student and professional groups with emphasis now on Society of Automotive Engineers as a Faculty Advisory to the Student Branch and with the design of Mini Baja Vehicles. He is also active with the Western Canadian Industry Exhibition Association.

ORAM, Guy

Position:

M.Sc. Student
Dept. Mechanical Engineering
Queen's University
Kingston, Ontario
Canada K7L 3N6

Citizenship:

Canadian and British

Qualifications:

B.Sc. (Eng.) 1986, University of Dublin,
Republic of Ireland
M.Sc. Mech. Eng. 1991, Queen's University,
Kingston, Canada

Experience:

During the period 1987 to 1989, Mr. Oram was a Research Engineer with Armstrong Research Corporation of Toronto. From 1989 to 1991, Mr. Oram was a Teaching and Research Assistant in the Department of Mechanical Engineering at Queen's University.

OSEI, DANKYI N.

Position:

Graduate student, Faculty of Engineering,
University of Regina, Regina, Saskatchewan,
CANADA.

Country and year of Birth:

Ghana, 1960

Qualifications:

B.Sc. Geological Eng. , U.S.T Kumasi-Ghana
Member, G.S.G.

Experience:

From 1988 to 1991, Mr. Osei was employed at the Water Resources Research Institute of the Council for Scientific and Industrial Research, Accra, Ghana. In 1991, he enrolled at the University of Regina for his Master degree in Civil Engineering, specializing in groundwater modelling. He is the author of one and co-author of five scientific and technical papers.

PATENAUDE, Pauline B.

Position:

Marketing Specialist, Equipment and Machinery Development Division, Saskatchewan Research Council, Saskatoon, Saskatchewan, Canada, S7N 2X8

Qualifications:

B.Comm., 1987, U of S.
Associate Member, Canadian Institute of Marketing.
Professional Member, American Marketing Association.

Experience:

Ms. Patenaude's career is focused on providing management, marketing and financial consulting services. She has acted as Marketing and Investments Manager for High Line Group, International Sales and Marketing Manager for Saskatoon Fresh Pack Foods Ltd., and Marketing Specialist for the College of Agriculture, University of Saskatchewan. As well as her duties at S.R.C., Ms. Patenaude is currently completing an M.B.A. at the University of Saskatchewan.

PAWLAK, Wanda

Position:

Research Officer, Alberta Research Council, P.O. Bag 1310, Devon, Alberta, Canada.

Experience:

Dr. Wanda Pawlak holds a Ph.D. degree in chemistry and technology of coal. From 1970 to 1981 she was a member of the academic staff at the University of Poznan, Poland. In 1981 she joined the University of Alberta as a post-doctoral Fellow. Since 1983 she has been with the Alberta Research Council. Her research interests are in the areas of chemistry and structure of coal matter, beneficiation and transportation of coal.

PAYANDEH, Shahram

Position:

Assistant Professor, School of Engineering Science, Simon Fraser University, Burnaby, British Columbia, V5A 1S6

Country and Year of Birth:

Iran, 1957

Qualifications:

B.S. Mech. Eng. 1981, U. of Akron (Ohio)

M.S. Mech. Eng. 1982, U. of Akron

M.A.Sc. Mech. Eng. 1986, U. of Toronto

Ph.D. Mech. Eng. 1990, U. of Toronto

P.Eng. (Ontario)

P.Eng. (British Columbia)

Associate Member, ASME

Member, IEEE

Experience:

From 1980 to 1982, Dr. Payandeh was a *research associate* in the rotor dynamics group of University of Akron. In 1982 he was a *consultant engineer* in Sault Ste. Marie (Ontario). From 1983 to 1984 he was a *graduate research assistant* at the Robotics and Automation Laboratory of the University of Toronto. From May of 1984 to 1986 he was a *research engineer* in the Center for Flexible Manufacturing at McMaster University. From 1986 to 1990 he was a *graduate research assistant* at the Robotics and Automation Laboratory of the University of Toronto. In January of 1991 he joined the School of Engineering Science of Simon Fraser University where he is currently an *Assistant Professor*. Dr. Payandeh's areas of research are: grasping using dexterous mechanical end-effectors, robot mechanics and control, the kinematics of manipulating mechanisms and knowledge-based approaches in robotics.

PELLEY, Charles W.

Position:

Assistant Professor, Department of Mining Engineering, Kingston, Ontario, Canada, K7L 4V2

Qualifications:

B.Sc. Geology (Hons.) 1968, Memorial University of Nfld.

M.Eng. Mining 1973, McGill University

P.Eng. (Ontario)

Experience:

From 1968 to 1971 Mr. Pelley served as a military Engineer. From 1973 to 1978 he worked for Asbestos Corporation in Thetford Mines, P.Q., in both open pit and underground operating positions. In 1978 and 1979 he was underground superintendent with Esso Minerals at their Gays River Mine. From 1979 to 1984 he worked for the Iron Ore Company of Canada, Carol Operations, for three years as chief of the long range planning group. Mr. Pelley joined Queen's in late 1984.

PRAKASH, Avneesh

Position:

Graduate Student, Industrial Systems Engineering, University of Regina, Regina, Saskatchewan, Canada S4S0A2.

Country and Year of Birth:

India, 1968

Qualification:

B. Tech. Mech. Eng. 1991, IIT, Kanpur, India

Research Interests:

Mr. Prakash is a student in the master of science programme in Industrial Systems Engineering at the University of Regina. His areas of interest are plasticity, modeling and simulation of manufacturing systems and robotics.

Z. F. Sang

Position

Associate professor. Faculty of Mechanical Engineering Department, Nanjing Institute of Chemical Technology. Nanjing, 210009 P. R. China.

Country and Year of Birth

P. R. China, 1942

Professional Experience:

Graduated in 1965 from Department of Mechanical Engineering, Nanjing Institute of Chemical Technology. From 1965~1966, as a design engineer in Nanjing Chemical Engineering Co., Design and management of process equipment and pressure vessels. 1966~1986, as a lecturer, senior lecturer, Department of Mechanical Engineering, Nanjing Institute of Chemical Tech. 1986~1988, as a visiting scholar, Research associate, Department of Mechanical Engineering, University of Illinois at Chicago, pursuing study and research on analysis and design of pressure vessels. 1989~1992. Associate professor, Department of M. E., Nanjing Institute of Chemical Tech.

Field of Research and teaching:

Structure and strength of pressure vessels and piping. Analysis and design of process equipment.

Professional Memberships:

ASME member.

Member of Chemistry and Chemical Engineering Association of China.

Member of Mechanical Engineering Association of China.

Publication:

He is an author or Co-author of over 20 scientific and technology papers on analysis, calculation, and experiment for pressure vessels and piping system.

SCOULDING, David

Position:

Manager, Thermal Power, Monenco Inc., Calgary, Alberta, Canada. T2P 3W3

Country and Year of Birth:

England, 1942

Qualifications:

B.Sc. Eng. 1965, Imperial College, U. of London
M.Eng. Controls 1975, Concordia U.
P.Eng. (Alberta)

Experience:

Mr. Scoulding has 25 years' experience in utility, refinery, petrochemical and industrial related projects in the area of project and technical management, instrumentation and control systems and mechanical design. He has participated in the design of more than 15 thermal power units ranging in size from 50 to 400 MW. He is responsible for projects and studies involving advanced technologies for the generation of power and steam in an environmental acceptable manner.

SESHADRI, RANGASWAMY

Position:

Professor and Dean
Faculty of Engineering
University of Regina,
Regina, Saskatchewan,
Canada S4S 0A2.

Country and Year of Birth:

India, 1945.

Qualification:

B. Eng. (Hons)	Mech. Eng., 1967	U. of Jabalpur
M. Tech.	Eng. Mechanics,	Indian Inst. of Tech.
	1969	Madras, India
M. Sc.	Mech. Eng., 1971	U. of Calgary
Ph. D.	Mech. Eng., 1973	U. of Calgary

Experience

Dr. Seshadri worked as Project Engineer with EBA Engineering Consultants in Edmonton between 1974 and 1977. During this period, he was involved in numerous geothermal design problems of Northern Canada. In 1977 he moved to Associated Engineering Services Ltd. in Edmonton, where he worked as a Mechanical Design Engineer in petrochemical related projects. In 1978 he joined Syncrude Canada in their Design Engineering Division. In his capacity as a Senior Mechanical Engineer with Syncrude's Corporate Development in Edmonton, Dr. Seshadri has worked on a number of design engineering problems relating to Capital and Plant Support activities in the areas of Upgrading and Utilities. Between 1981 and 1986 he worked in the Syncrude Plant in Fort McMurray as an Engineering Associate. In 1987 he moved to the University of Regina where he is currently Dean of Engineering. His current research interests are in the area of elevated temperature component design and robust design methods. Dr. Seshadri has published over fifty technical articles in Canadian as well as International Journals, primarily in the areas of mechanics and is the author of a book.

SHEK, C. H.

Position:

Research student, Department of Mechanical Engineering, University of Hong Kong, Pokfulam Road, Hong Kong.

Country and Year of Birth:

Hong Kong, 1966

Qualifications:

B.Sc. Mechanical Engineering, 1988, University of Hong Kong.

Experience:

Mr. Shek is a research student jointly supervised by Drs. Lai and Duggan since 1989.

SHERIFF, Alpha Y.

Position:

Research Assistant
Dept. of Mechanical Engineering
Queen's University
Kingston, Ontario
Canada K7L 3N6

Country and Year of Birth:

Sierra Leone, 1956

Qualifications:

B.Eng. (Honours) Mech. Eng. 1982, Fourah Bay College, University of Sierra Leone
M.Sc. Mech. Eng. 1987, Queen's University, Kingston, Canada

Experience:

From 1983 to 1985 Mr. Sheriff was employed by Sierra Leone National Power Authority. He has been working as a Research and Teaching Assistant at Queen's University from 1985 to the present.

SINGH, M.C.

Position:

Professor
Department of Mechanical Engineering
The University of Calgary
Calgary, Alberta, Canada
T2N 1N4

Qualifications:

B.Sc., Punjab University, 1952
M.Sc. (Structures), University of Minnesota, 1956
Ph.D. (Applied Mechanics), University of Minnesota, 1962

With the Faculty of Engineering, Department of Mechanical Engineering, University of Calgary since July 1968.

Research Interests:

Stress Analysis; Vibrations of linear, non-linear and Symmetric Systems; Similarity Analysis with Applications to Linear and Non-Linear Wave Propagation; Developments in Thermoelastic and Thermoplastic Constitutive Relations; Wave Propagation in Non-Linear Thermoelastic and Inelastic Solids.

Shahab Sokhansanj

Position

Professor of Agricultural and Bioresource Engineering, University of Saskatchewan, Saskatoon, S7N 0W0

Country and Year of Birth

Iran, 1943

Qualifications

B.Sc. Agricultural Engineering, University of Tehran, 1966
M.Sc. and Ph.D. Michigan State University, 1977
P.Eng. Saskatchewan

Experience

Worked for 1977 to 1979 as a research fellow at the University of Minnesota, U.S.A. Prior to graduate studies, he spent seven years with agricultural production and equipment manufacturing firms as design and production engineer.

Dr. Sokhansanj is the immediate past Chairman of the grain and feed processing committee of the American Society of Agricultural Engineers. He is active in farm equipment standardization projects in Canada and the U.S.A., serving as member or chairman on several national and international committees. He is a registered Professional Engineer and is consulted extensively by equipment manufacturers in Canada.

Dr. Sokhansanj has authored or co-authored more than 100 scientific papers and books related to crop processing. In 1986, he received a certificate from the Richard B. Russell Agricultural Research Center of the U.S. Department of Agriculture for his innovative research in the area of dielectric properties of grains. In 1991, the Society of Agricultural Engineering recognized his outstanding contribution to research and teaching by awarding him the 1991 John C. Clark Award.

SZYMOCHA, Kazimierz

Position:

Associate Research Officer, Alberta Research Council, P.O. Bag #1310, Devon, Alberta, Canada.

Country and Year of Birth:

Poland, 1942

Qualifications:

M.Sc. Mech. Eng. 1969, Technical University of Wroclaw, Poland
Ph.D. Mech. Eng. 1975, Technical University of Wroclaw, Poland.
Member of International Freight Pipeline Society

Experience:

Dr. Szymocha holds a M.Sc. and Ph.D. in mechanical engineering from the Technical University of Wroclaw. In 1969 he joined the Technical University of Wroclaw. In 1981 he spent a half year of sabbatical study at École Nationale Supérieure al Electricité et de Méchanique, Polytechnique de Lorraine, Nancy, France. In 1982, Dr. Szymocha visited Canada and spent two years as a Post-doctoral Fellow at the University of Alberta and Visiting Scientist at the Alberta Research Council. After visiting Canada, he returned to Poland to work as an Associate Professor in the Faculty of Engineering at the Technical University of Wroclaw. Dr. Szymocha joined the Alberta Research Council in 1988 and has made significant contributions to coal upgrading and soil clean-up programs. He is the author and co-author of 43 publications in Refereed Journals or Conference Proceedings.

TABARROK, Bez

Position:

Professor and Chair, Department of Mechanical Engineering, University of Victoria, Victoria, BC, Canada V8W 3P6

Country and Year of Birth:

Iran, 1938

Qualifications:

B.Sc. Mechanical Eng. 1963, Wolverhampton Polytechnic
D.Phil. Structural Dynamics 1965, Oxford
P.Eng. (Ontario)
Fellow, CSME, EIC

Experience:

Dr. Tabarrok came to Canada from U.K. in 1965, as a post-doctoral fellow. He was then appointed as an assistant professor of Mechanical Eng's in 1966. Later he was prompted to associate and full professor of Mechanical Engineering in University of Toronto and in 1987 he joined the university of Victoria to start up a new Department of Mechanical Engineering.

Dr. Tabarrok's research has been in the area of computational mechanics and he has provided consulting services to numerous engineering companies on problems of stress, vibration and heat transfer analysis.

TABOUN, Salem M.

Position:

Assistant Professor, Department of Mechanical Engineering, University of Ottawa, Ottawa, Ontario, Canada. K1N 6N5

Country and Year of Birth:

Tripoli, Libya, 1953

Qualifications:

B.Sc. Mech. & Ind. Eng., 1977, U. of Tripoli
M.Sc. Industrial Eng., 1981, U. of Miami
Ph.D. Industrial Eng., 1986, U. of Windsor
Senior member of IIE and SME

Experience:

From 1987 to 1989, Dr. Taboun held an Assistant Professor in Industrial Systems Engineering at the University of Regina. Currently, he is an Assistant Professor in the Department of Mechanical Engineering at The University of Ottawa. He has supervised several M.A.Sc. students in Manufacturing and Industrial Ergonomics, and published more than 30 articles in International journals and conference proceedings.

TICHLER, PAUL R.

Position:

Section Leader
Reactor Safety Evaluation Group
Reactor Division
Brookhaven National Laboratory
Upton, New York 11973
U.S.A.

Country and Year of Birth:

U.S.A., 1931

Qualifications:

B.Ch.E, City College of New York
M.S.Ch.E., Columbia University, New York
P. Eng. (New York State)

Experience:

1958-Present, Brookhaven National Laboratory
1988-Present, Section Leader of Reactor Safety Evaluation.
1965-1988, Provided technical support for operating research reactors in areas of reactor safety including accident analysis, thermal hydraulics, reactor physics and water chemistry.
1958-1965, Member of the High Flux Beam Reactor design staff in areas of thermal hydraulics and water chemistry.

TONG, Liyong

Position:

Postdoctoral research associate, Department of Mechanical Engineering, University of Victoria, Victoria, BC, Canada
V8W 3P6

Country and Year of Birth:

China, 1963

Qualifications:

B.Eng. Eng. Mech. 1982, Dalian Institute of Technology
M.Eng. Computational. Mech. 1985, Dalian Institute of Tech.
Ph.D. Composite Materials and Structures. 1988, Beijing U. of Aeronautics and Astronautics

Experience:

From 1988 to 1990, Dr. Tong was appointed Director in the Division of Applied Mechanics at Beijing Star Center of Technology. Since Nov. 1990, he is a postdoctoral research associate in the Department of Mechanical Engineering at University of Victoria. He is the author or co-author of over 30 scientific and technical papers.

TONTIWACHWUTHIKUL, Paitoon

Position:

Assistant Professor, Faculty of Engineering, University of Regina, Regina, Saskatchewan, Canada.

Country and Year of Birth:

Thailand, 1961

Qualifications:

B.Eng. Chem. Eng. 1983, King Mongkut's Inst. Tech.
B.A. Political Sci. 1983, Ramkhamhueng Univ.
M.Eng. Chem. Eng. 1986, U. of British Columbia
Ph.D. Chem. Eng. 1991, U. of British Columbia

Experience:

After obtaining Ph.D. degree in Chemical Engineering, Dr. Tontiwachwuthikul joined the Pulp and Paper Research Centre (UBC-PAPRICAN) as a Postdoctoral Fellow (supported by the Networks of Centres of Excellence). In July 1991, he was appointed as an Assistant Professor in the Faculty of Engineering, University of Regina. Dr. Tontiwachwuthikul is active in the research areas of gas processing technologies, air pollution control engineering, CO₂ recovery and utilization, pulping processes, and expert systems in engineering.

TORVI, Allan A.

Position:

Associate Professor, Department of Mechanical Engineering, The University of Calgary, Calgary, Alberta, Canada

Country and Year of Birth:

Canada, 1936

Qualifications:

B.E.Sc., 1963, University of Western Ontario
M.E.Sc., 1966, University of Western Ontario
P.Eng. (Alberta)
Fellow, EIC

Experience:

From 1954 to 1958, Professor Torvi served his machinist's apprenticeship with INCO's mines mechanical department. He worked as a vibrations analyst at United Aircraft of Canada Ltd. (now Pratt and Whitney Canada Ltd.) in Longueuil, Quebec, during the 1964-69 period. He joined The University of Calgary in July, 1969, as an Assistant Professor and has held administrative positions at The University of Calgary as Acting Head of Common Curriculum, Assistant Dean, Director of Engineering Programs (Continuing Education), and is currently Acting Associate Head (Manufacturing).

TRAN, David, V.D.

Position:

Research Scientist
Combustion Dynamics Ltd.
Medicine Hat, Alberta, Canada
T1A 8B5

Country and Year of Birth:

Viet Nam, 1957

Qualifications:

B.Sc., Mech. Eng. 1986, University of Calgary
M.Sc., Mech. Eng. 1990, University of Calgary

Experience:

From 1986 to 1991, I worked at the University of Calgary as Research Associate under Professor M.C. Singh. At that time, our topics included wave propagation in nonlinear elastic solids, nonlinear thermoelastic solids and non-conducting nonlinear materials. Currently, I am working at Combustion Dynamics Ltd. directly related to DRES (Defence Research Establishment Suffield) as Research Scientist. There I have studied and developed numerical modelling of underwater explosion including one and two interfaces.

VIRARAGHAVAN, Thiruvinkatachari

Position:

Professor, Faculty of Engineering,
University of Regina, Regina, Saskatchewan,
Canada, S4S 0A2

Country and Year of Birth:

India, 1934

Qualifications:

B.E. Civil Eng., 1955, University of
Madras
M.Sc. Public Health Eng., 1963, University
of Madras
Ph.D. Civil Eng., 1975, University of
Ottawa
P. Eng. (Saskatchewan, New Brunswick,
Ontario)
Fellow, CSCE; Fellow, ASCE; Fellow, IWEM
(UK); Fellow, Inst. Engineers (India).

Experience:

From 1955 to 1965, Dr. Viraraghavan
was employed as an Assistant Public Health
Engineer with the Government of Tamilnadu,
Madras, India. During 1965 - 1970, he was
an Assistant Adviser in Public Health
Engineering, Ministry of Works and Housing,
New Delhi, India. He worked as a senior
environmental engineer with ADI Limited,
Consulting Engineers, Fredericton, New
Brunswick during 1975 - 1982. He joined
the Faculty of Engineering, University of
Regina in 1982. He is a member on the
editorial board of four journals and is the
author or co-author of more than 200
publications.

WANG, Tsunkuei

Position:

Professor, Department of Applied Mechanics,
Beijing University of Aeronautics and
Astronautics, Beijing, P.R. China.

Country and Year of Birth:

China

Qualifications:

B.Sc. Mathematics, Peking University
M.Eng. and M.Aero., California Institute of
Tech.
Ph.D Stanford University.

Experience:

At Stanford, Dr. Wang was a student of
Timoshenko, and at Caltech he worked with
Von Karman. After returning to China in
1952, Dr. Wang became a professor in the
Department of Flight Vehicle Design and
Applied Mechanics at Beijing University of
Aeronautics and Astronautics. He is the
president of the Chinese Society of
Composite Materials and the editor-in-chief
of ACTA Materialae Compositae SINICA.

WEAVER, D.S.

Position:

Professor, Dept. of Mechanical Engineering,
McMaster University

Qualifications:

B.A.Sc., University of Toronto
M.A.Sc., University of Toronto
Ph.D., University of Waterloo
Fellow, ASME
American Academy of Mechanics

Experience:

Dr. Weaver taught in the Department of Civil Engineering at the University of Waterloo before coming to McMaster in 1971. His current teaching includes courses in solid mechanics, vibrations, mechanical design and failure analysis. His research interests are generally in the area of flow-induced vibrations with applications to problems in heat exchangers, bellows expansion joints hydraulic structures and pipeline acoustics. Dr. Weaver has served as Chair of the Department of Mechanical Engineering for six years, and as Associate Dean of Engineering (External Relations) for three years. He is currently chairman of the NSERC Mechanical Engineering Grant Selection Committee.

WRIGHT, Ralph D.

Position: Self Employed

Country and Year of Birth:

Canada 1927

Qualifications:

P.Eng. Civil Engineering 1961, Alberta
P/O RCAF Permanent Force (1963)

Experience:

From 1955 to 1963, Mr. Wright was employed by the Bridge Branch, Department of Highways, Saskatchewan. In 1963 he joined the RCAF and worked on engineering studies and projects in Alberta, British Columbia, Manitoba, Ontario, Quebec, and Nova Scotia. From 1970 to 1977 he worked for the Department of Highways in northern Alberta. Since 1977 he has been self employed in structural design and municipal engineering in Alberta and Saskatchewan. He also took on an assignment in Nicaragua and in Lesotho, Africa and continues to work on the design of agricultural equipment for farmers in developing countries. He also works on harnessing energy from wind, solar, stream flow and draft animals.

YU, Shu Dong

Position:

Ph.D. Student, Department of Mechanical Engineering, University of Toronto, Toronto, Ontario, Canada M5A 1A4.

Country and Year of Birth

China, 1962

Qualifications:

B.Sc. Mech. Eng. 1982, Jiangxi Metallurgical Institute
M.Sc. Appl. Mech. 1984, Northeast U. of Technology
Member - CSME; Student Member - ASME, AIAA, CACSMA

Experience:

From 1984 to 1988, Mr. Yu taught in the Department of Applied Mechanics at Northeast University of Technology, specializing in structural dynamics. In 1989, Mr. Yu spent one year of research leave at the University of Ottawa, Ottawa, Ontario. He has published over 25 technical papers.

Y.Z.Zhu

Position:

Lecturer, Mechanical Engineering Department Assistant Dean, Director of Central Lab. Nanjing Institute of Chemical Technology, Nanjing 210009, P. R. China

Country and Year of Birth:

P. R. China, 1958

Qualifications:

B.Sc. Mech.Eng.1982,Nanjing Institute of Chem. Tech.
M.Sc. Mech.Eng.1992,Nanjing Institute of Chem. Tech.
Member, CMES

Experience:

From 1982 to 1987, primary lecturer, and from 1987 to now is a senior lecturer of mechanical engineering department, Nanjing Institute of Chemical Technology. Field of research and teaching is stress analysis of structure and pressure vessel. In 1987, gained the third prize of advanced scientific technology for the research on experimental equipment of U-shape bellows expansion joints by Chemical Engineering Ministry of China, and is the author or Co-author of over 10 scientific and technology papers on analysis, calculation and experiment for pressure vessels and piping system.

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University of Regina



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**11e SYMPOSIUM
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**ADDENDUM
PROCEEDINGS**

"ENGINEERING PLANT LIFE ASSESSMENT"

M.A. IWANIW and S.D. Bhole

EDITORS

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Life Extension and Component Condition Assessment in the United States

R. B. Dooley, W.P. McNaughton⁺ and R. Viswanathan

Exploratory and Applied Research Division
Electric Power Research Institute
Palo Alto, California U.S.A.

⁺Daedalus Associates, Inc.
Mountain View, California U.S.A.

ABSTRACT

In the United States, the speed of adoption of a technology for the assessment of key fossil-fuel power plant components is influenced by a number of technical and non-technical factors. This paper addresses the current status of assessment philosophies, creep damage detection tools, and for condition assessment of specific components. Utility presentations and a survey of present practice are used to quantify the use of methods in the U.S. Three specific power plant components are discussed: boiler tubes, main steam/hot reheat piping, and high temperature headers. For each component, a list of common problems, a recommended approach for assessment and the extent of "penetration" of the optimal technology into common application are provided.

The results of this study show that multi-level and phased assessments have seen widespread adoption in U.S. utilities. Of the various methods for creep detection, only replication (for voids) is in general application by utilities or their contractors; although there is increasing use of alternatives. Creep crack growth analysis methods are applied to determine remaining life and set reinspection intervals for cracked components. Analysis of boiler tube failures directly by U.S. utilities is widespread and increasing in its effectiveness. Evaluation of remaining tube life using oxide scale techniques is popular and increasing in its application. Multi-level approaches are generally applied to the other two components. U.S. utilities perform themselves part, or in a few cases all, of the required activities, consistent with their particular expertise in inspection, metallurgy, analysis and lifetime prediction.

INTRODUCTION AND BACKGROUND

Introduction

The analysis of aging fossil-fired power plants for life extension and the application of component assessment methodologies in the U.S. varies considerably from utility to utility. The penetration into the industry of key assessment techniques and methods was investigated through a formal survey of eighteen utilities and by analysis of status presented by an additional sixteen U.S. utilities. The principal focus of results reported here is the evaluation of high-temperature components operating at or near the creep regime. However, the stage is set by some general comments about the overall approach to life extension and condition assessment in aging units. This is followed by an overview of available tools for the detection of creep damage, and by a discussion of specific components and key assessment items; notably for boiler tubes, main steam and hot reheat piping, and high temperature headers.

Background

In the U.S., assessment of aging fossil-fired power plants in the early to mid-1980s often involved major outages (three month minimum length) during which the condition of literally hundreds of pieces of plant equipment were examined, many through the use of expensive (and often redundant) sampling programs. Often there was no balance in the approach with very detailed evaluations of one component and little or no evaluation of equally critical components elsewhere in the unit.

As a result of the shortcomings in such methods a program was developed to formalize and optimize for U.S. utilities an approach to be taken for life extension. After the development of generic guidelines (1), a demonstration project (2) was put in place with ten U.S. and Canadian utilities to test the general applicability of the methods. The general approach for an overall life optimization strategy, shown in Figure 1, has been extensively documented. Central to the assessment of particular equipment is a progressive, multi-level approach, shown in Figure 2, a philosophy pioneered at the CEGB for the evaluation of high temperature headers (3). Table 1 shows the defining characteristics of each of three levels of analysis.

The multi-step approach to the assessment has evolved and has been adopted by most U.S. utilities. The approach is modified when common sense dictates that activities generally applied later in the sequence need to be moved to earlier roles. For example, it is often expeditious to identify critical locations or to determine allowable flaw sizes analytically before embarking on an inspection program (Level II) even though such activities are generally considered to be a part of the most detailed (Level III) evaluations. Such "pre-analysis" is most clearly advantageous when the analysis of inspection indications is complex and likely to take longer than that allowed by the constraints of the outage, or where knowing the critical damage level can trigger a replace or repair option.

As a result of the use of phased methods and multi-level equipment assessment, life extension activities have been matched to normal utility outage with concurrent cost savings. The following case study from a demonstration project participant illustrates the point.

The first unit evaluation by this participant required sixteen weeks. The original schedule was for a twelve week outage; however, an additional four weeks were taken to make in-situ repairs of defective conditions found during the inspections. Since the initial unit evaluation, the average time to complete an entire unit (except for turbine work which is matched to planned turbine outages) has declined to four weeks or less. With twelve units to evaluate, the savings in outage required for completion of baseline studies has been projected to be more than 48 weeks. Background work (Level I) is performed during operation and prior to the outage. The four week evaluation time also does not include some of the equipment common to multiple units, such as coal handling systems. The improved ability to conduct unit evaluations can be attributed to the clear definition of work to be performed largely the result of the preparatory background (Level I) work, and to good outage planning and coordination.

Savings from application of these methodologies have been estimated to be \$7.7 million. This is based on an outage time savings of 48 weeks, average unit size of 275 MW, an assumption that the average capacity factor for units inspected is 70%, and an average replacement power differential of \$5/MW-hr.

The progressive assessment methodology, the need for integrated management and planning, and the use of phased approaches has been adopted and refined internationally. In one recent international conference (4) presentations from fourteen different countries reported the use of systems that were conceptually similar to those outlined above.

Methods for Assessing Creep Damage

Methods to detect developing creep damage and to predict remaining life of high temperature components once creep cracks initiate have been under extensive development during the past few years. Viswanathan and Gehl (5) have reviewed advances in life assessment techniques for these components including methods for detection of creep damage and for creep crack growth. Table 2A overviews the available methods including open issues; techniques in bold print were specifically polled in the survey. Table 2B provides a list of the applications and some additional notes for the key techniques.

Individual Component Assessment

In addition to surveying for general approaches to life extension and for details of creep damage detection, utilities were polled on their application of life assessment to a number of key plant components. The results for three specific high-temperature components are discussed in this paper: boiler tubes, main steam and hot reheat piping, and high temperature headers. A discussion of common problems and recommended approach for each is found in the next three sections to set the stage for the discussion of survey results.

ANALYSIS OF BOILER TUBES

Despite an overall improvement in the availability of major plant equipment in the last decade, boiler tube failures (BTF) still rank as the number one equipment problem in fossil power plants in the U.S. Currently BTF represent around 3-4% of lost availability (6); in order of availability loss the leading BTF mechanisms are:

- Corrosion fatigue
- Flyash erosion
- Hydrogen damage
- High temperature creep
- Short term overheating
- Sootblower erosion
- Waterwall fireside corrosion
- Falling slag erosion

This ranking is believed to be accurate for the entire U.S. fossil industry although developed from a sample of sixteen utilities. Similar rankings have been recorded in many other countries worldwide (6). Of most interest for the topic of this conference is analysis of high temperature creep, the fourth mechanism in the ranking. This category actually includes several separate and distinct mechanisms with similar failure modes, of which the two primary ones are:

- Fireside corrosion caused by low melting point alkali iron trisulphates that form on the tubing as a result of corrosive coals. Flue gas induced erosion may also play a part in removing protective oxides. As a result, tube hoop stress is increased and failure eventually results from long-term creep.
- Long-term overheating of a particular tube section caused by poor initial design and incorrect alloy choice. The tube section oxidizes at a higher rate in the sulfur-containing environment and flow-induced tube flats are formed where tube failure eventually occurs. The most common location for this type of failure is just before a transition to a higher grade of material.

It is important to determine the root cause of a high temperature creep failure since the solutions depend upon cause. It is likely that in North America that long term overheating is the more common mechanism of the two listed above as coals are very infrequently corrosive enough to cause fireside corrosion, unlike the situation in the United Kingdom for instance (7).

During the past twenty years a number of techniques have been developed to assess remaining life of superheater/reheater tubing subject to high temperature creep. A recent survey of 44 utilities determined the percent using each of the common methods (6):

- Estimating temperature by measuring internal oxide thickness (43%).
- Wall loss/corrosion rate determination by measuring wall thickness with ultrasonics. At least two measurements are required separated by a number of years (34%).
- Measuring tube temperatures (infrequently used because of expense and burn out of thermocouples) (25%).
- Estimating temperature by material carbide degradation or hardness (25%).
- Statistical approaches (e.g., Weibull analysis) (16%).

- Uniaxial testing (14%).
- Tube burst testing (2%).

As can be seen from these results (and confirmed by our survey), the method of choice is currently one based on oxide scale analysis. Figure 3 provides an overview of the approach. The method, procedures, and present understanding of the limitations of the methodology are described in detail in Viswanathan, et. al. (8). A summary of the method is included here to illustrate key issues.

Tubes or tube assemblies at risk, those with thickest steamside oxide and thinnest wall thickness, are identified (Steps 1, 2, and 3 of Figure 3). Procedures for stubs, dissimilar metal welds (DMWs) and tubing (Step 4 in the figure) have been developed.

For tube analysis, remaining life of each tube is predicted by a computer code (EPRI-developed TUBELIFE, or available commercial codes from U.S. vendors such as NOTIS, TUBECALC, TUBEPRO, etc). Sacrificial tube samples are then removed from select locations and subjected to laboratory metallurgical analysis and/or isostress rupture testing so as to refine the remaining life estimates.

As an example, the computer code (TUBELIFE) uses present oxide thickness and an isothermal oxide growth rate to determine a current "equivalent" temperature of the tube. An effective stress is calculated from the tube dimensions. The temperature and stress values are then extrapolated back to initial conditions assuming linear growth of steam-scale and fire-side wastage along with heat-transfer properties of the steel and oxide scale. The average metal temperature and stress for every 1000 hr interval from beginning of service is then calculated iteratively. For each interval, oxide scale thickness and wall thinning are estimated under nonisothermal conditions. The life expended at each 1000 hr interval is computed from the temperature increase caused by the steam side oxide, the stress increase caused by fireside corrosion, and stress-rupture curve at each calculated temperature. Life fractions are summed to determine total life consumed. Remaining life for each tube is then calculated.

Available codes for oxide scale analysis cover various materials depending upon the vendor and version available. For example, early versions of TUBELIFE available through EPRI, and well used by survey participants, evaluated only ASME SA213-T22. The most recent version (Spring, 1992 release date) now covers a total of ten materials: T1, T2, T5, T9, T11, T12, T21, T22, SA178, and TP304. Included in the new release will be code modifications to help interpret lifetimes for tubes that have been chemically cleaned.

ANALYSIS OF MAIN STEAM AND HOT REHEAT PIPING

These piping components are mostly produced from low-carbon, low-alloy steels, P11 ($1\frac{1}{2}\text{Cr}-\frac{1}{2}\text{Mo}$) and P22 ($2\frac{1}{2}\text{Cr}-1\text{Mo}$). On rare occasion austenitic stainless steels are employed for the highest temperatures and pressures, such as are characteristic of supercritical units. Pipe is usually extruded (seamless), although piping produced from formed, seam-welded plate has been used in many hot-reheat lines. Damage in high temperature piping systems is often the result of creep, fatigue and microstructural changes. Susceptible locations include both welds (longitudinal and girth orientations) and base material.

Seam-weld Damage

A U.S. industry wide inspection program was initiated following a 1985 longitudinal rupture in a 32-inch diameter, seam-welded steam line at Southern California Edison's Mohave Station. A rupture in a seam welded hot reheat line also occurred in 1986 at Detroit Edison's Monroe Station. Following the Mohave failure, inspection programs were initiated by a large number of utilities to establish the suitability of seam-welded piping for continued safe operation. Initial criteria for piping at risk were:

- $1\frac{1}{2}\text{Cr}-\frac{1}{2}\text{Mo}$ and $2\frac{1}{2}\text{Cr}-1\text{Mo}$ piping designed between 1952 and 1967 to operate at or near the stress limits of ANSI B31.1 Code.
- Presence of longitudinal seam welds in pipes carrying steam at temperatures over 950°F for $1\frac{1}{2}\text{Cr}-\frac{1}{2}\text{Mo}$ and 1000°F for $2\frac{1}{2}\text{Cr}-1\text{Mo}$.
- Use of submerged arc welding to manufacture pipes.

Following review of industry data, further criteria for high risk locations have been established. These include:

- Locations of weld repair.
- Details of the flux type used during welding and the distribution and type of inclusions present in the finished weld.

EPRI performed a survey of utilities and the results have been published for fifteen units (9). Two units exhibited no NDE indications, surface cracks or creep damage. Five units contained apparent fabrication defects that were detected by ultrasonic examination. Four of these indications were evaluated metallographically by removal of plug samples and were found to be hot tears, slag or porosity. No creep damage or crack extension was associated with these defects. One indication had not been removed, but appeared to be centered in weld metal. One clamshell-welded 2½Cr-1Mo elbow was found by surface replication to contain microcracks in the heat-affected zone. Similar cracking was observed on a transverse section. No creep cavitation was found, and it appeared that this cracking resulted from welding.

The remaining seven units exhibited crack growth, creep damage or both, in longitudinal welds. They represented both 2½Cr-1Mo and 1½Cr-½Mo, high single pass heat input and low multiple pass heat input, straight pipe spools and clamshell elbows, and more than one piping manufacturer. In several cases the damage was found to have extended along several feet of weldment.

Guidelines for inspection and evaluation of seam welded steam piping have been issued. They contain: a review of the failures and flaws found to-date; industry experiences in performing examinations; details of the research and development supporting the guidelines; review procedures; replication and NDE procedures; methods for evaluating NDE and replication results; procedures for remaining life estimates; and the bases for establishing re-inspection intervals. A overview of the method can be found in Viswanathan, Dooley and Saxena (10).

Girth-weld Damage

Damage in girth welds can be oriented both circumferentially and transversely to the weld. Transverse cracking or creep damage is caused by long term application of pressure stress. Circumferential creep cracking, however, is primarily caused by bending and system stresses, due to supports and hangers not operating properly. These bending or axial tensile systems stresses will

typically result in crack development at the site of the weakest weldment microstructure. Damage of this form is often observed in the partially transformed region of the HAZ where the welding thermal cycles are sufficient to coarsen the carbides present thus leading to a narrow band of low strength material. Fatigue can interact with the creep damage mechanism in either direction, and the extent of the fatigue damage will be dependent on how the plant is operated. Fatigue damage is also influenced by geometric stress concentration, such as the local weld geometry or backing bars. Although bad supports can be fixed and plant operation can be modified, long term creep under a steady hoop stress eventually leads to problems.

Base Metal Damage

Two base metal damage mechanisms of concern for high energy piping are thermal softening (degradation of the microstructure with coarsening and agglomeration of the carbides) and creep cavitation damage. These damage mechanisms have been discussed by Cane (11) and by Parker and Sidey (12).

Assessment of High Energy Piping

A general phased approach to the evaluation of high energy piping has been developed. The recommended assessment is carried out in sequential phases of increasing complexity until an acceptable inspection interval is predicted or until the economic advantage of replacement is apparent. Table 3 illustrates the various requirements of this three level approach. Computer codes (such as the EPRI-sponsored SLIC and vendor-developed PCPIPE) provide fracture mechanics calculation of creep crack growth. Reference (10) reviews the methods developed for analysis of both creep crack initiation and creep crack growth.

ANALYSIS OF HIGH-TEMPERATURE HEADERS

Most high-temperature headers in U.S. units are constructed from $1\frac{1}{2}\text{Cr}-\frac{1}{2}\text{Mo}$ or $2\frac{1}{2}\text{Cr}-1\text{Mo}$ steels. The principal types of damage encountered in high temperature headers are creep, fatigue, creep-fatigue and microstructural changes (thermal degradation) coupled with a possible environmental effect from steamside oxidation. The damage mechanisms are manifested by cracking in a variety of forms, depending on location and metallurgical structure. These have been classified in detail in (13) from U.S. and U.K. experience with

inspection of headers; the results are summarized in Table 4.

Base Metal Damage

Three broad categories of base metal damage have been identified:

- Cracks have been observed at the outside and/or inside surfaces of the outlet tee nozzle. This damage typically occurs close to the axis of the header in the axial direction. At the outside surface, damage appears to be related to creep driven by multi axial loads and cracking can develop without measurable creep strain. At the inside surface, damage appears to be related to fatigue.
- Damage may be developed in the base metal as a result of localized deformation. This damage is detectable by measurement of component strain and is caused by local high temperatures. Swelling may be accompanied by cracking; however, life may be limited by the total strain or the rate of strain accumulation.
- Cracking can ensue at the inside surface of bore holes. Cracks may initiate in a star-burst pattern with no apparent preferred orientation, however, propagation may lead to a dominant crack. Dominant cracks are usually observed to follow a principal stress direction (axial or circumferential).

Ligament-Damage

A survey of 376 high temperature headers, specifically inspected for ligament cracking has been performed. The results of this survey are summarized in Table 5. A total of 157 secondary superheater outlet headers were examined in this survey; 28% were found to have ligament cracking. Of the other 219 high temperature headers examined, only 3% were found to contain this form of damage.

Further analysis of the data from the superheater outlet headers was carried out to consider the age of the header and the alloy of construction. Review of the data obtained to date reveals that no simple trend exists between cracking and header age or alloy type. However, examination of the incidence of cracking for the typical ligament geometries indicates that "pie-shaped" ligaments which terminate at a point at the inside surface appear to be at high risk of crack development. Cracks generally initiate in the tube bore hole and are oriented parallel to the axis of the tube hole. Link-up of cracks between holes on the inside surface then leads to propagation to form cross ligament cracks. In other situations, propagation along the bore hole

may be more extensive and in extreme situations this form of damage can lead to leaks at stub tubes. In this latter case life may be governed by failure rate at the tubes.

Weld Damage

Cracking associated with welds has also been observed, at stub tube welds, header end-cap welds, saddle welds, girth welds and longitudinal seam welds. Damage is primarily caused by creep, but the creep process may be accelerated by fatigue. Depending upon the local conditions, geometry and microstructure, damage may be developed in the weld metal, at the fusion line or in the heat affected zone. Evaluation of the specific location and morphology of cracks can play an important role in identifying the cause of damage, and thus permit accurate corrective measures. The tendency for a specific crack orientation can be evaluated by consideration of component stresses and strains.

Condition Assessment for High Temperature Headers

The success of life assessment methods is to a large extent linked to the ability to locate and sample potentially damaged material. In this respect, unless the failure mode can be accurately identified, the location for material removal or examination is not obvious. The use of instrumentation, to assist this process, e.g. thermocouples, strain gauging, etc. has increased. As a general observation, microstructural degradation, leading to strain accumulation, occurs in the component shell, whereas creep cracking is generally associated with weldments. Cracks at tube-hole ligaments may be related to creep, fatigue, or some combination of these mechanisms.

The scheme for header life assessment is based on CEGB practice (3) and derivatives of that practice (14). A Level III assessment (in the absence of obvious cracks) would typically consist of the following elements:

- A refined stress analysis by finite-element methods to identify and quantify the highly-stressed regions.
- Detailed inspection of the header by dye penetrant, magnetic particle, UT, and/or radiographic methods, especially at welds and penetrations.
- Sampling of the header material for accelerated creep-rupture testing to establish the actual properties of the material.

- Replication of the surfaces in high stress, high temperature regions so that high-resolution microscopy can be used to ensure that creep damage has not accumulated locally to the degree that microscopic cavities are visible or that the microstructure has not otherwise degenerated. Replication results can also supplement other remaining life methods (15).
- Metallurgical evaluations based upon isostress creep analysis of miniature specimens (16).

In the presence of cracks, methods for predicting creep crack and creep-fatigue crack growth have been developed. A review of available methodologies can be found in Viswanathan and Gehl (5). Evaluating the seriousness of the crack(s) and establishing reinspection intervals for those defects which are left in service is performed through application of crack growth analyses.

SURVEY METHOD

The preceding sections have outlined the status of developed methods for the assessment of key components. The balance of this paper examines the status of the application of these methods in U.S. utilities.

Thirty-four utilities were canvassed for input to this question. Two data collection efforts were used to obtain information: a detailed survey was sent to eighteen U.S. utilities and sixteen additional utilities provided presentations of the status of their boiler component assessment programs. Of the eighteen utilities that were formally surveyed, ten had been previous participants in a life extension demonstration project, five were participants in a Boiler Tube Failure reduction project, and three were chosen because of an expressed long-standing interest in life extension.

RESULTS

The results of the survey are divided into three areas of interest:

- Overall concepts of equipment assessment
- Use of specific techniques for creep damage detection
- Approaches to specific equipment

Overall Approaches to Equipment Condition Assessment

Table 6 lists the responses to a number of the general questions surveyed. The following specific conclusions come from survey results:

1. There continues to be a strong interest in condition assessment of equipment and the extended use of older units in U.S. utilities.
2. All surveyed utilities have initiated some form of program, and in fact, all participants have scheduled the inspection of critical components.
3. There is an almost universal use of phased inspections and refurbishments.
4. There is an almost universal use of multi-levels of condition assessment; although the evaluation of electrical equipment and balance of plant are less likely to employ such methods.
5. Almost all of the respondents used outside contractors to perform major equipment inspections and remaining life analyses. This was particularly true for the evaluation of mechanical equipment and balance of plant. It was less true for electrical equipment, for which both inspections and remaining life analysis were "seldom" done by outside contractors.
6. When using outside contractors, the respondents "always" or "often" specified the use of multi-level assessment methods. When not specifically requested, such multi-level assessments were almost universally employed by outside contractors anyway.
7. Only about 60% of the utilities surveyed have written statements of operating objectives for their units; only about half have formal procedures to determine root cause of equipment failures, although for mechanical equipment the percentage was slightly higher.

One trend which has developed in the use of multi-level methods, and was cited on some of the responses is that most users have found that most equipment assessment begins with Level II methods. Generally because of poor historical information, it has not been possible to make equipment decisions based on the detail of Level I. Invariably, at least baseline inspections (Level II), including visual, magnetic particle, ultrasonic testing, etc. have been required. At the same time there is clear evidence that the prior practice of extensive sampling of material by destructive means is not used; selective material sampling is required and is used, but only in a reasoned way as part of Level III methods.

Creep Detection Methods

Table 7 shows the results of the survey for creep detection methods, which are consistent with the authors opinions of the wider practice in the U.S. utility industry. The methods cited are ranked in the table according to percent of respondents that use the method in-house "often". Thus replication is used "often" in-house by 87% of the respondents. The question "Access?" meant, "Do the personnel in your organization have general access to this tool?". "Contractor use" was meant to determine whether a utility's contractor had used the stated tool to help evaluate the utility's equipment.

Specific conclusions are:

1. U.S. utilities have access to most of the creep analysis methods either in-house or through contractors. However, replication (particularly for voids) is really the only technique to see widespread use by utilities in-house (that is, applied by utility personnel directly instead of using a contractor). It is also the only method to see almost universal availability through contractors to the respondents.
2. None of the rest of the creep detection methods was as widely applied by utilities in their own analysis, although all of the listed methods have seen some application as indicated in Table 7. In the sixteen oral utility presentations, hardness-based techniques were clearly the second favorite method to replication, and a significantly higher number of users was evident than in the written survey respondents.
3. A fill-in portion of the written survey queried utility respondents to comment on needed tools for assessment and several commented on the need for NDE methods for creep detection. The sixteen utilities that provided presentations were also clearly interested in improving their understanding and application of the available methods.

Application to Individual Components

Table 8 displays the responses from the survey and from the qualitative presentations. The format is similar to that for the previous table. The utilities were first asked about whether they had access to the technology (Yes/No) and then the extent to which they used that particular tool or method. Finally, if they use contractors, whether the contractor used that tool was queried. The number of responses to the last question was small, in most cases where respondents did not supply a reply it is assumed that the respondent did not have knowledge of contractor use. It should also be noted

that where a method is surveyed (such as oxide scale analysis of boiler tubes) the percentage of contractors applying the method, and the total number of responses is high, whereas, when asked about a particular (often EPRI-sponsored) tool, such as the TUBELIFE program, both the number of positive responses and the total number of responses is lower. This is a natural result of the fact that there are a number of commercial suppliers of most of the specific tools, each with an individual version.

Boiler Tubes. Preventing boiler tube failures has been a major emphasis during the past five years. The message that tube failures are preventable and unacceptable for most failure mechanisms is widespread among utility respondents. As shown in Table 8, almost all respondents to the survey list the Boiler Tube Failure Manual as a "often" used tool. A similar response was noted in utility presentations. The Manual includes technical information about identifying failures and preventative measures.

The use of the oxide scale method for analysis of remaining life, particularly the availability and use of the TUBELIFE program (an EPRI-sponsored oxide scale program) both in-house at utilities and through contractors was widespread. This was confirmed in utility presentations. The low level of access to the TUBELIFE program (only 40% of respondents) may reflect it's relatively recent vintage. The fact that 75% of those with access cite it as being used often is an indication of its application. It is expected that new program versions covering additional materials will increase the penetration of the technology. One major problem cited by several utilities in presentations was the amount of scatter in predictions of life among the various applications of the oxide technique. As has been noted above, this continues to be an area of active consideration.

An extensive methodology on dissimilar metal welds was also reasonably widespread in its use. Among those with access to the methods and program, about 25% indicated that they did not use the tools.

The Interim Consensus Guidelines on Cycle Chemistry, a recommended tool to help prevent water chemistry-related tube failures saw fairly high marks for frequent use.

High Temperature Headers. The methods outlined earlier in this paper for high temperature headers are fairly frequently used in house by the survey respondents. The utility presentations indicated that for headers, and for other major equipment, the issue is not generally whether they perform an entire evaluation on their own, but to what portions (inspections, metallurgical examinations, stress analyses, remaining life calculations, etc.) do their existing resources allow them to execute in-house.

Few utilities could confirm satisfactory remaining life of either headers or steam piping using only Level I methods. Nearly all have performed inspections (Level II), mostly using techniques such as visual, dimensional measure, ultrasonic testing, etc to measure for dimensional changes (e.g., swelling) or cracking. This has been supplemented by the application of some creep initiation detection methods such as replication for voids and hardness testing. Most have thus established baseline condition in their headers.

If defects are found, a decision is made whether to repair immediately or continue to run the header. In either case, reinspection intervals are set to monitor the progress of the damage. To make run/repair decisions and to set inspection intervals, crack growth and fracture mechanics analyses are used (Level III). Utilities will utilize in-house programs or more often use the services of a specialty contractor to provide such evaluations. The particulars of most of these methods tend to be proprietary to the contractor. Material sampling is specified whenever a more precise estimate of mechanical properties is required for the assessment, such as when the assumption of most conservative properties does not allow for sufficient accuracy in the analysis.

Replacement for cause (rather than after an arbitrary period of operation) has become standard procedure for many utilities. For example, a surveyed utility had ordered a replacement header based on the number of hours of operation, extent of previous repairs, and detection of moderate creep damage. During a repair, material samples were pulled from the most seriously affected areas of the header and accelerated creep tests were performed. The material showed sufficient remaining life that a decision to cancel the header order and continue to operate with the existing component was made.

Utilities surveyed had found a full range of defect types and sizes and had applied the range of resolutions. Minor defects were often left as-is, or ground out as noted above. More significant defects had been weld-repaired including the appropriate pre- or post-weld heat treatment. In some cases, the most significant defects may require replacement of part or all of a header. However, few headers have been replaced in the U.S. relative to the incidence of cracking presented in Table 5.

Main Steam and Hot Reheat Piping. These systems have certainly had major attention and deservedly so because of the safety concern, particularly with seam-welded pipe. Most utilities with seamed reheat lines have inspected for obvious defects.

As with header analysis discussed above, once defects are detected in main steam or hot reheat piping, an analysis of significance is completed, often in conjunction with an outside contractor. Material sampling is used when uncertainty in properties affects the decision about how to proceed with the resolution of the defect. The Guidelines have seen some use directly by utilities, the associated fracture mechanics codes less use. This is consistent with the position of most utilities in directing the type of work to be performed, but, particularly for specialized technical skills such as fracture mechanics evaluations, using outside contractors. That such contractors usually have their own proprietary codes, is consistent with the relatively low level of usage of EPRI-developed generic codes such as the one shown for steam piping (SLIC).

Several utilities surveyed replaced seamed hot reheat lines with seamless construction. In a few special circumstances, replacement has been made without comprehensive inspection or remaining life analysis. In one case, the utility had compared the cost of inspection, and the probable outcome based on industry risk factors with the cost of replacing the line and opted for replacement. In a second case, seamed pipe was found in several units in short (28') sections and a decision to replace without analysis was determined to be the lowest cost option.

DISCUSSION OF RESULTS AND IMPLICATIONS FOR TECHNOLOGY DIFFUSION

Table 9 reviews the general capabilities required for life assessment of major components. Within the survey, the various demonstration projects discussed, and the presentations reviewed, there was a range of utility capability. At least six of the smaller utilities in the data base had virtually no in-house capability, which naturally limits their own use of the methods. At the other end of the spectrum are large, well-equipped utilities that not only have all the listed capabilities but often in-house research and development efforts that produce proprietary tools for such equipment evaluations.

Most U.S. utilities have some in-house inspection capability (generally of standard methods such as PT, MT, UT and sometimes RT), although this is not universal. As noted in the discussion of creep methods above about three quarters perform some type of replication with somewhat fewer using other creep damage detection methods. Refined stress analysis and fracture mechanics is performed by outside contractors for many utilities.

In general, the more of these capabilities that an individual utility has, the more likely that they will have adopted a comprehensive scheme for equipment evaluation. Which technologies are adopted for use in-house depend directly on whether a particular utility has personnel knowledgeable in that discipline and the equipment to perform the analysis. With the exception of a few, large, resource-rich utilities, most U.S. utilities still depend upon a range of contractor-supplied methods.

Most U.S. utilities seem to be moving away from reactive programs (such as the response to steam line problems following field failures) to more preventive programs, although there is still a tendency to follow industry experience. Recent financial constraints have also preempted a number of programs that might have offered long-term benefits to individual units. Also, a number of respondents reported that decision-making for the choice of methods and equipment to be analyzed rests with the individual plant managers who may or may not have access to a complete range of assessment tools.

Two paths to successful diffusion of technology are evident in the results of this survey work. First, a sponsoring research organization like EPRI can fund the development of, and offer to members, tools that can be directly

applied by as many utility users as possible. A good example of this in the work discussed above is the analysis and prevention of boiler tube failures. The methods, supporting documents and procedural changes have become very widespread, although not universal among U.S. utilities. The second path to diffusion of methods is to sponsor research that eventually results in the development of tools by an active group of third parties: specialty contractors, architect/engineers and original equipment manufacturers. The early sponsorship of oxide-scale techniques for creep-life assessment of tubes by EPRI is such an example from the work described above. Although vendor developments have paralleled, EPRI-sponsored Code versions, the method has seen considerable adoption throughout the industry in a relatively short time. It is expected that based on the survey a similar rapid diffusion of creep detection methods will occur during the next few years.

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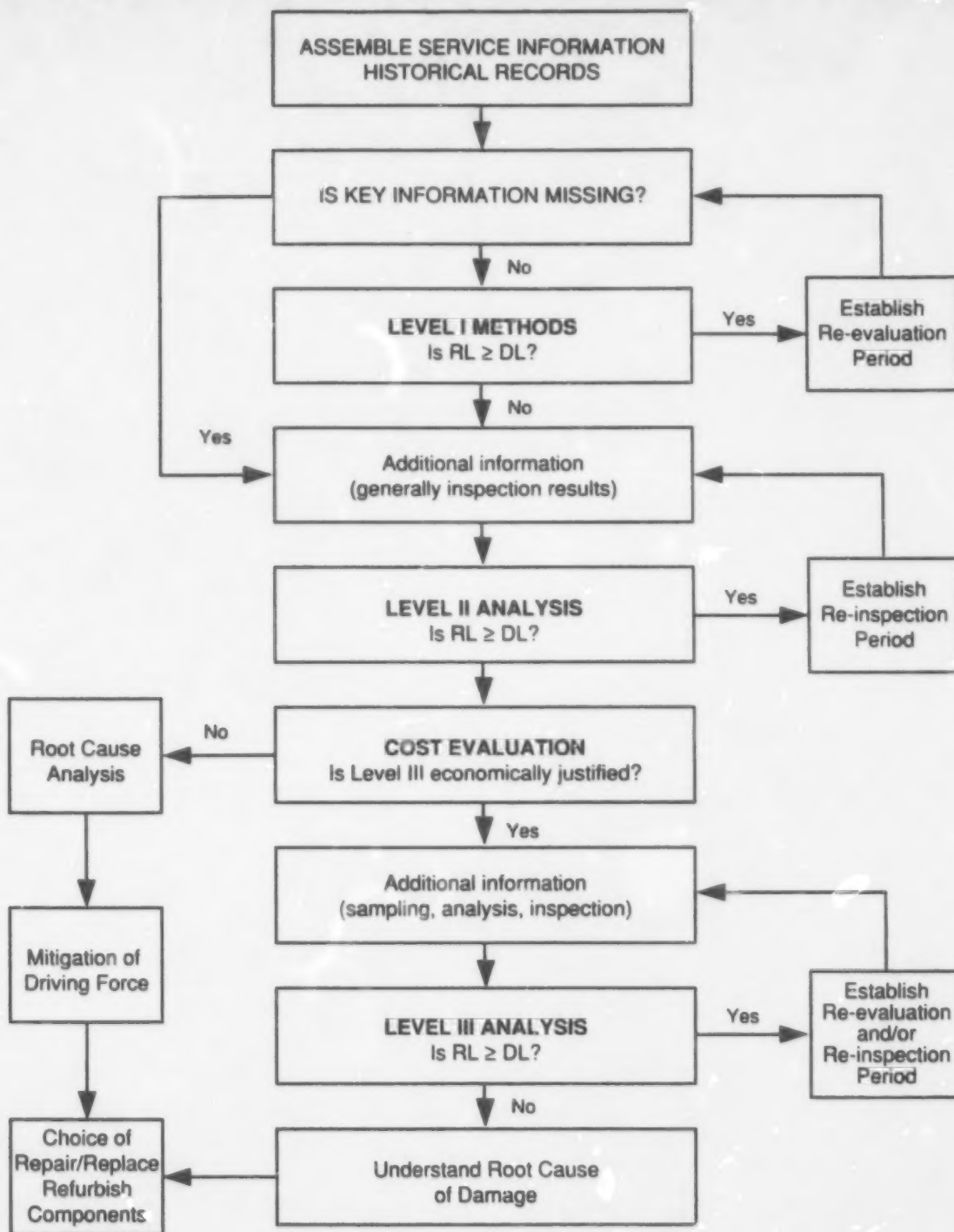
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RL: Remaining Life of Component Predicted by Evaluation
DL: Desired Life of Component (Derived from Unit Objectives)

Re-inspection or re-evaluation periods: Chosen by evaluation of results of analysis, sensitivity to measurements, critically of component, lead-time, safety factors, etc.

Figure 2. KEY STAGES OF MULTI-LEVEL COMPONENT ASSESSMENT

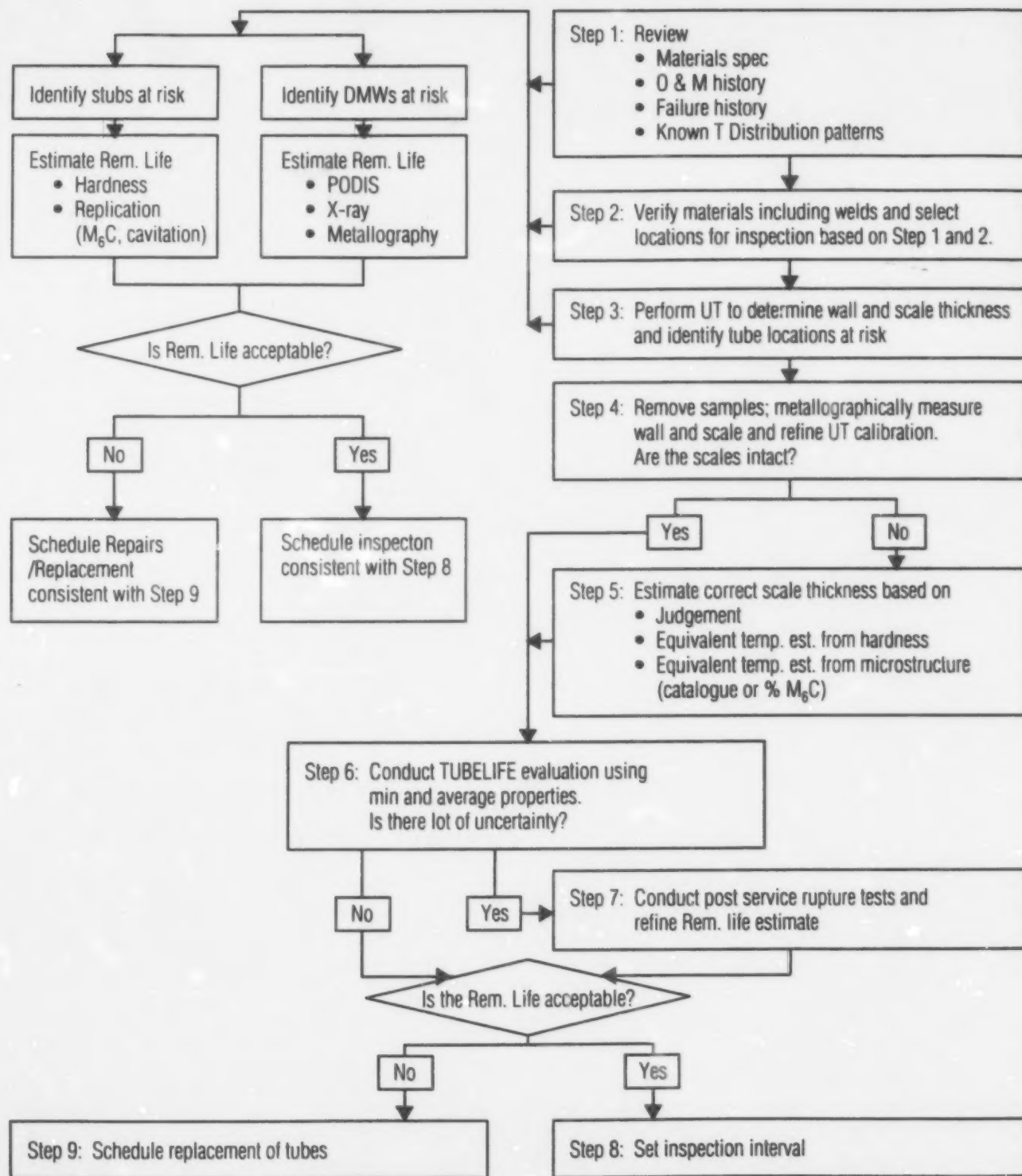


Figure 3. ROAD MAP FOR REMAINING LIFE ASSESSMENT OF SUPERHEATER/REHEATER TUBES

TABLE 1

DATA REQUIREMENTS FOR A MULTI-LEVEL APPROACH
MECHANICAL EQUIPMENT

FEATURE	LEVEL I	LEVEL II	LEVEL III
Failure History	Plant Records	Plant Records	Plant Records
Dimensions	Design or Nominal	Measured or Nominal	Measured
Condition	Records or Nominal	Inspection	Detail Inspection
Temperature and Pressure	Design or Operational	Operational or Measured	Measured
Stresses	Design or Operational	Simple Calculation	Refined Analysis
Material Properties	Minimum	Minimum	Actual Material
Material Samples Required?	No	No	Yes

Successive Levels Invoke:

More Rigorous Assessment

More Accurate Operation Data Required

More Accurate Estimate of Equipment Remaining Life

TABLE 2A

LIFE ASSESSMENT TECHNIQUES AND THEIR LIMITATIONS FOR CREEP DAMAGE

----- Crack Initiation -----		----- Crack Propagation -----	
Technique	Issues		
Calculation	Inaccurate		
Extrapolation of past experience	Inaccurate		
Conventional NDE	Inadequate resolution		<i>Technique:</i> Analysis combines NDE, stress analysis and crack growth with an end-of-life criterion to predict remaining life.
High-resolution NDE:			
Acoustic emission			
Positron annihilation			
Barkhausen noise analysis			
Strain (dimension) measurement			
Rupture testing	Not sufficiently developed at this time		<i>Issues for Crack Propagation Methods:</i>
	Uncertainty regarding original dimensions		Uncertainties in NDE results
	Lack of clear-cut failure criteria		Lack of adequate crack growth data in creep and creep-fatigue
	Difficulty in detecting localized damage		Lack of a clear-cut end-of-life criterion under creep conditions
	Difficulty in sample removal		Difficulty in assessing toughness of in-service components
	Difficulty in using as a monitoring technique		
	Validity of life-fraction rule		
	Effects of oxidation and specimen size		
	Uniaxial-to-multiaxial correlations		

TABLE 2A
(continued)

LIFE ASSESSMENT TECHNIQUES AND THEIR LIMITATIONS FOR CREEP DAMAGE
(Crack Initiation and Crack Propagation)

----- Crack Initiation -----	----- Crack Propagation -----
Technique	Issues
Microstructural evaluation	Quantitative relationships with remaining life are lacking
Cavitation measurement	
Carbide-coarsening measurements	
Lattice parameter	
Ferrite chemistry analysis	
Hardness monitoring	
Oxide scale measurements for tubes	Kinetics of hot-corrosion and constant-damage curves
Replication strain monitoring	

Bold represents methods specifically addressed by survey.

Adapted from Viswanathan and Dooley (17).

TABLE 2B
CREEP DETECTION AND ASSESSMENT TECHNIQUES:
APPLICATIONS AND ADDITIONAL NOTES

Technique	Typical Applications	Notes
Replication	Header stubs Girth welds (weld and HAZ)	Estimates of creep damage only.
Replication, "A" parameter	High stress regions in base metal, Weld HAZ	Estimates of creep damage only.
Carbide spacing	Header, steam pipe base metal	Probable use as a monitor of temperature or with other methods for life assessment.
Accelerated rupture testing	Header OD base metal Steam pipe base metal	Creep damage only. Need oxidation correction.
Mini-specimen creep testing	Similar when limited material available	
Hardness	Base metals	Currently limited to estimates of average temperature, not remaining life.
Replica strain monitoring	Welds, HAZ, Local strain regions of interest	Used to establish strain rate by re-examination at periodic intervals.
Ultrasonic testing	Long seam welds Ligaments (for long cracks) Locations with extensive creep damage	
Crack growth Analysis	Wherever cracks or crack-like defects are found during inspections	Creep crack growth and creep-fatigue crack growth methods, material properties available.

TABLE 3

THREE LEVEL APPROACH TO CONDITION ASSESSMENT OF HIGH ENERGY PIPING

LEVEL I - PRELIMINARY PIPING SYSTEM EVALUATION

- Review fabrication, construction, operating history
- Perform hot piping walkdown and survey
- Preliminary stress analysis
- Preliminary life fraction analysis
- (Preliminary creep crack growth)
- Recommend examination plan

LEVEL II - PIPING INSPECTION

- Perform cold piping walkdown and survey
- Examine to identify seam welds
- MT examination of welds
- UT scan at high sensitivity
- Resolve weld toe indications
- Map and size all long continuous or intermittent indications
- Perform hardness and dimensional measurements
- Perform alloy analysis
- Take replicas of suspect areas

LEVEL III - ENGINEERING ANALYSIS

- Removal of material sample
- Estimate creep rate mismatch; calculation fusion boundary stress concentration
- Estimate number and location of susceptible weld beads
- Estimate inclusion density
- Calculate time to crack initiation, or current size of defects
- Calculate crack size versus operating time
- Recommend inspection interval
- Elastic/creep piping stress analysis

TABLE 4
LONG TERM DAMAGE IN ELEVATED TEMPERATURE HEADERS

<u>Position</u>	<u>Damage Mechanism</u>
Branch Connection Saddle	Creep Cavitation
Header Body Swelling	Thermal Softening
Longitudinal Seam Weld and HAZ Cracking	Creep Cavitation
Girth Butt Weld and HAZ Cracking	Creep Cavitation
Stub Tube HAZ Cracking	Creep Cavitation
Bore Hole Circumferential Ligament Cracking	Thermal Fatigue and Oxide Initiation

Source: Ellis, et al. (13)

TABLE 5

HEADER INSPECTION RESULTS BY HEADER TYPE
(Evaluated for Ligament Cracking)

<u>Header Type</u>	<u>Number Inspected</u>	<u>Number with Tube Bore Cracks</u>	<u>Percentage</u>
Secondary Superheater Outlet Headers			
1½ Cr Material	73	26	36%
2½ Cr Material	76	17	22%
Operating Temp. > 1050°F (565°C)	<u>14</u>	<u>6</u>	<u>43%</u>
Total	157	44	28%
Reheat Superheater Outlet Headers	118	2	2%
All Other Headers	101	4	4%

TABLE 6

OVERALL APPROACHES TO CONDITION ASSESSMENT

Question/Issue	Responses				
	Increased	40%	Decreased	10%	Same 50%
1. During the past two years, has the emphasis on evaluation and use of older units in your system?					
2. Has a schedule for inspection of critical components been established?	Yes	100%	No	0%	
3. Were (or will) the inspections all be conducted during a single outage?	Yes	13%	No	87%	
4. If a baseline study and subsequent inspection identified major equipment (rotors, headers) that needed replacement or refurbishment has					
i. That refurbishment been scheduled?	Yes	100%	No	0%	
ii. Has all refurbishment been completed?	Yes	20%	No	80%	
5. Does your utility have a written statement of operating objectives for all units?	Yes	60%	No	30%	Some 10%
6. If failures occur to mechanical, electrical or balance of plant equipment, is there an established, formal procedure to determine the root cause of failure?					
i. Mechanical Equipment	Yes	70%	No	20%	Unknown 10%
ii. Electrical Equipment	Yes	50%	No	30%	Unknown 20%
iii. Balance of Plant	Yes	40%	No	30%	Unknown 30%

TABLE 6

OVERALL APPROACHES TO CONDITION ASSESSMENT
(continued)

Question/Issue	Always	Often	Responses		Unknown
			Seldom	Never	
7. If your company does some or all of its own evaluation of equipment (inspections, remaining life analysis, stress analysis, etc.) is a progressive or multi-level approach used					
i. Mechanical Equipment	22%	78%	0%	0%	0%
ii. Electrical Equipment	11%	50%	22%	0%	11%
iii. Balance of Plant	11%	50%	11%	0%	22%
8. To what extent does your company use outside groups (consultants, inspection companies, etc.) to perform equipment evaluations?					
Inspections					
i. Mechanical Equipment	10%	70%	20%	0%	0%
ii. Electrical Equipment	0%	11%	77%	0%	11%
iii. Balance of Plant	0%	66%	22%	0%	11%
Remaining Life Analyses					
i. Mechanical Equipment	10%	50%	30%	10%	0%
ii. Electrical Equipment	0%	11%	66%	22%	0%
iii. Balance of Plant	0%	22%	66%	0%	11%
9. If your company always or often uses others to assist in equipment evaluations, do you specify the use of multi-level methods in that work?					
i. Mechanical Equipment	12%	88%	0%	0%	0%
ii. Electrical Equipment	16%	50%	17%	0%	17%
iii. Balance of Plant	17%	33%	17%	0%	33%

TABLE 7

METHODS FOR DETECTING AND CHARACTERIZING CREEP-DAMAGED MATERIALS

Tool/Technique	Access?		----- In-house Use? -----			Contractor Use?	
	Yes	No	Often	Seldom	None	Yes	No
Replication, in general	89%	11%	87%	0%	13%	100%	0%
Oxide-scale analysis (for tubes)	86%	14%	50%	25%	25%	80%	20%
Accelerated Rupture Testing	75%	25%	33%	34%	33%	75%	25%
Replication, Carbide Spacing Analysis	40%	60%	33%	0%	67%	80%	20%
Replication Strain Monitoring	56%	44%	20%	0%	80%	60%	40%
Miniature Specimen Creep Testing	66%	34%	17%	50%	33%	71%	29%
Replication, "A" Parameter Methods	50%	50%	17%	0%	83%	60%	40%
Hardness-based Techniques	56%	44%	14%	28%	58%	83%	17%

TABLE 8

TOOLS AND TECHNIQUES FOR EVALUATING SPECIFIC COMPONENTS

Component	Tool/Technique (Application/Use)	Access?		----- In-house Use? -----				Contractor Use?	
		Yes	No	Often	Seldom	None	Unknown	Yes	No
BOILER TUBES	Manual for Investigation and Correction of Boiler Tube Failures	100%	0%	78%	11%	0%	11%	33%	66%
	Oxide-scale Analysis Methods (Analysis of Remaining Life)	75%	25%	67%	33%	0%	0%	80%	20%
	TUBELIFE (Computer Program) (Analysis of Remaining Life)	40%	60%	75%	25%	0%	0%	60%	40%
	Dissimilar Metal Welds (Seven volume reference on DMW failure, prediction and repair).	89%	11%	50%	25%	12%	12%	0%	100%
	PODIS (Computer Program) (Dissimilar Metal Weld Analysis)	44%	56%	50%	25%	25%	0%	0%	100%
CYCLE CHEMISTRY	Interim Consensus Guidelines on Plant Cycle Chemistry	89%	11%	63%	37%	0%	0%	50%	50%
HIGH TEMPERATURE HEADERS	Condition Assessment Guidelines for Fossil Power Plant Components (method shown in text)	100%	0%	50%	38%	12%	0%	33%	67%
MAIN STEAM AND HOT REHEAT PIPING	Guidelines for Steam Pipe Evaluations (Analysis and Inspection)	90%	10%	33%	66%	0%	0%	50%	50%
	SLIC (Computer Code) (Analysis of Pipe Cracking)	33%	67%	33%	33%	33%	0%	0%	100%

TABLE 9

CAPABILITIES LEADING TO SUCCESSFUL EXECUTION OF
COMPONENT CONDITION ASSESSMENT

Inspection capability

- Ability to specify what tests to run.
- Ability to determine what components to test.
- Ability to determine where to look on component.
- Have required equipment.
- Have ability to perform the inspection.
- Ability to interpret the results.

Metallurgical capability

- Knowledge of tests to perform.
- Have required equipment to analyze samples.
- For destructive samples, ability to minimize sampling scheme.
- Ability to interpret results.

Analysis capability

- Ability to set acceptable remaining life.
- Ability to calculate remaining life.
- Ability to set reinspection period.

Repair capability

- Ability to repair sampling locations if needed.

Financial/management capability

- Central engineering talent that can remain knowledgeable about changing technology.
- Presence of a strongly motivated, knowledgeable individual to direct activities (technology "champion").

ERRATA

Life Assessment of TransAlta Utilities' Wabamun Generating Station: A Status Report

B.D. Holter, TransAlta Utilities Corporation
J.S. Kestner, Monenco Inc
D. Scoulding, Monenco, Inc.

1. Under the Section "Execution", fifth paragraph, "...included the station,..." should be "included the **stator**,...".
2. Under the Section "Unit No. 3" second last paragraph, "...problems or drainage being found..." should be "...problems or **damage** being found...".

Analysis, Development & Testing of a
Tension Controlled Skyline Log-Yarding Carriage

C. Dayson
Tribology & Mechanics Laboratory
Institute for Mechanical Engineering
National Research Council
Vancouver, B.C.

1. In Figures 4 and 5, the illustrations should be interchanged.
2. Under the Section "Case Two: Capstan Slip Limit", the equation " $T_h < cF_s$ " should be followed by

$$" < 641 \text{ lbs. } (0.285 \times 10^4 \text{ N}) "$$

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